COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

November 1958

Experimental data for Yankee Atomic Electric Co plant comes from these tests at Westinghouse Electric Corp.

COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

Vol. 30 **No. 5**

November 1958

Feature Articles

Editorials

Departments

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BPA

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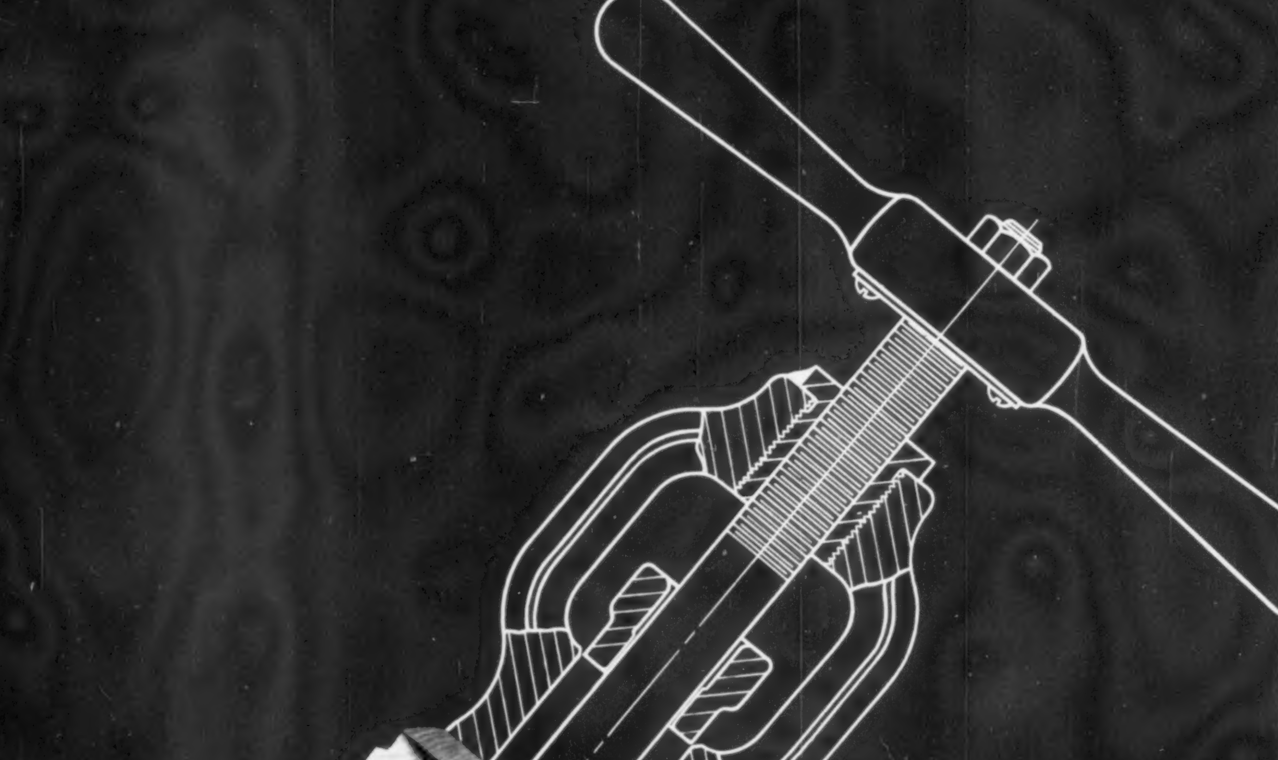


EDWARD UNIVALVE

What's New from Edward Valves, Inc.

New Products . . . Problems and Solutions. . . Information on Steel Valves from Edward, Long-Time Leader in the Field!

Shoulder, thread and weld make

Superior "Univalve" Body-Bonnet Joint

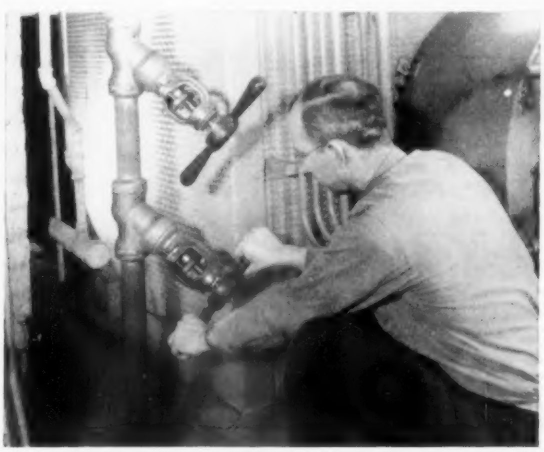
Edward valve researchers were assigned three main goals in developing the Univalve* . . . first, design a globe valve for high pressure-temperature service—that would *stay* leak-proof . . . second, eliminate—as far as possible—the necessity for maintenance . . . third, fabricate all pressure-containing parts of forged steel for maximum strength and soundness.

How well they succeeded is illustrated by this fact: in many hundreds of advanced temperature-pressure applications all over the world, Edward Univalves have given consistently superior service. The *reason* for this is simple . . . Univalves, properly maintained, simply do not leak.

WELD SEALS JOINT

In the Univalve, a bead of fine-grained weld seals the body-bonnet joint to maintain perfect pressure tightness in any service. A guiding section above the threads protects them from the seal-weld; the threaded section and body shoulder carry the pressure load and insure accurate alignment. The rugged threaded bonnet—with opening just large enough to accommodate the stem—provides a pressure-tight backseat. The radiused disk nut contacts the beveled bonnet backseating surface . . . isolates packing from line pressures and temperatures . . . stretches packing life.

While the Univalve rarely needs attention, even its tough integral Stellite seating surface *can* become scored under some conditions. To strip for inspection and possible re-lapping, the seal-weld is easily removed by machining or grinding or with carbon arc or oxy-acetylene scarfing tip. Besides simple disassembly and positive backseat advantages, Univalve's one-piece gland eliminates possible small parts loss during repacking.



IDEAL FOR BLOW-OFF SERVICE. Univalves meet ASME Code for blow-off service and are adaptable for all high pressure installations.

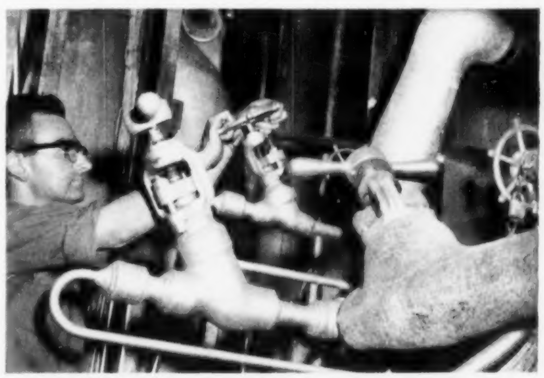
IMPORTANT UNIVALVE FEATURES:

Streamlined Flow Path reduces pressure drop, minimizes wear-producing turbulence. Univalve meets requirements for blow-off service.

Simple Packing Adjustment keeps packing maintenance down. Sturdy gland bolts, roomy yoke, one-piece forged gland.

Easy Open—Tight Close Operation of all Univalves 1¼" to 2½" made a reality with the exclusive Edward Impactor* handle.

Continuous Stellite Ring applied to body and disk, retains hardness under temperature and resists wear.



4500 LB UNIVALVE in service at Ohio Power Company's supercritical Philo station

*T.M. Reg. U.S. Pat. Off.

EDWARD VALVES, INC.

Subsidiary of ROCKWELL MANUFACTURING COMPANY

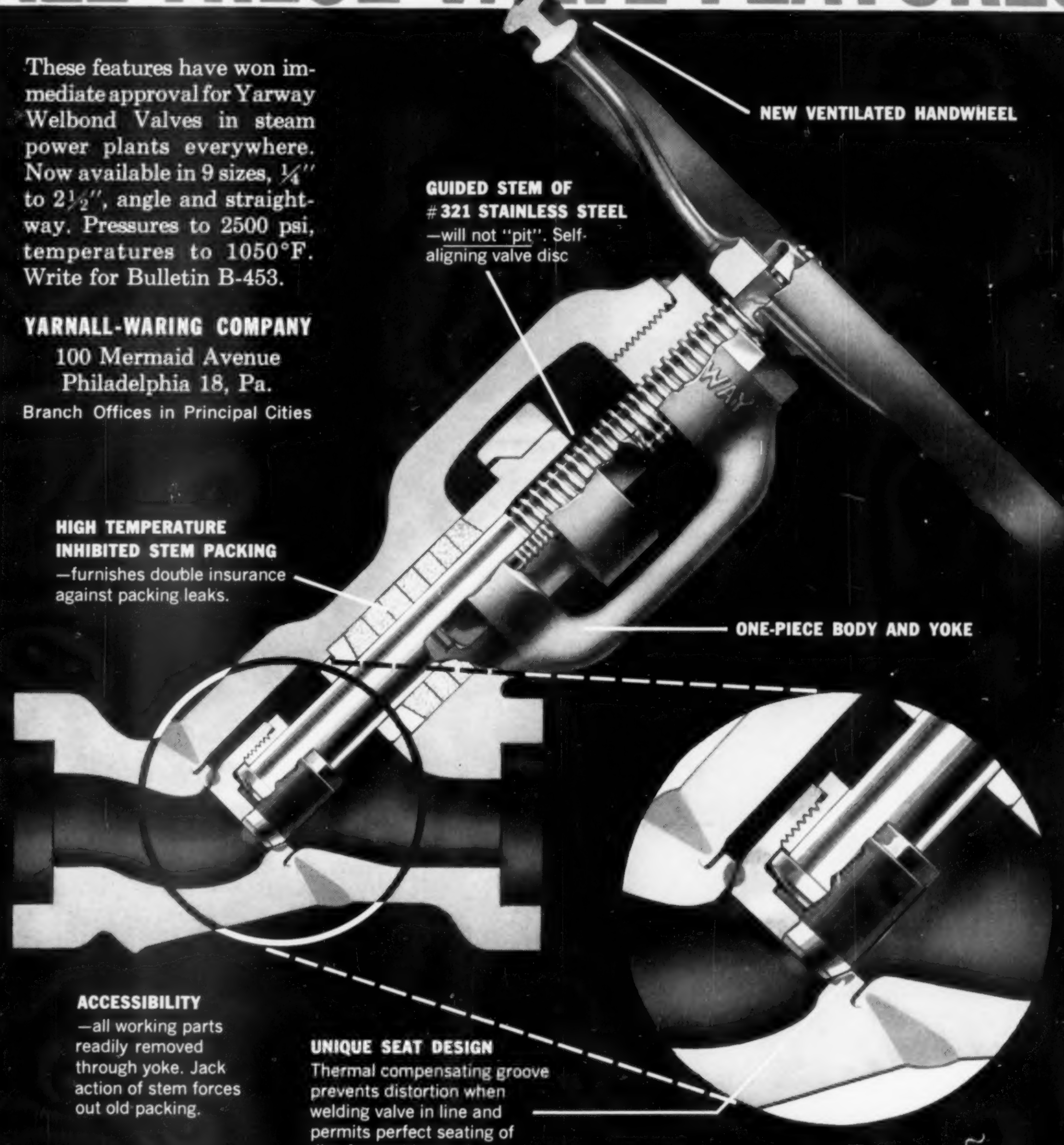
1206 West 145th Street, East Chicago, Indiana

Edward builds a complete line of forged and cast steel valves from ⅛" to 18"; in globe and angle stop, gate, non-return, check, blow-off, stop-check, relief, hydraulic, instrument, gage and special designs; for pressures up to 7500 lbs; with pressure-seal, bolted, union or welded bonnets; with screwed, welding or flanged ends.

ONLY WELBOND® COMBINES ALL THESE VALVE FEATURES

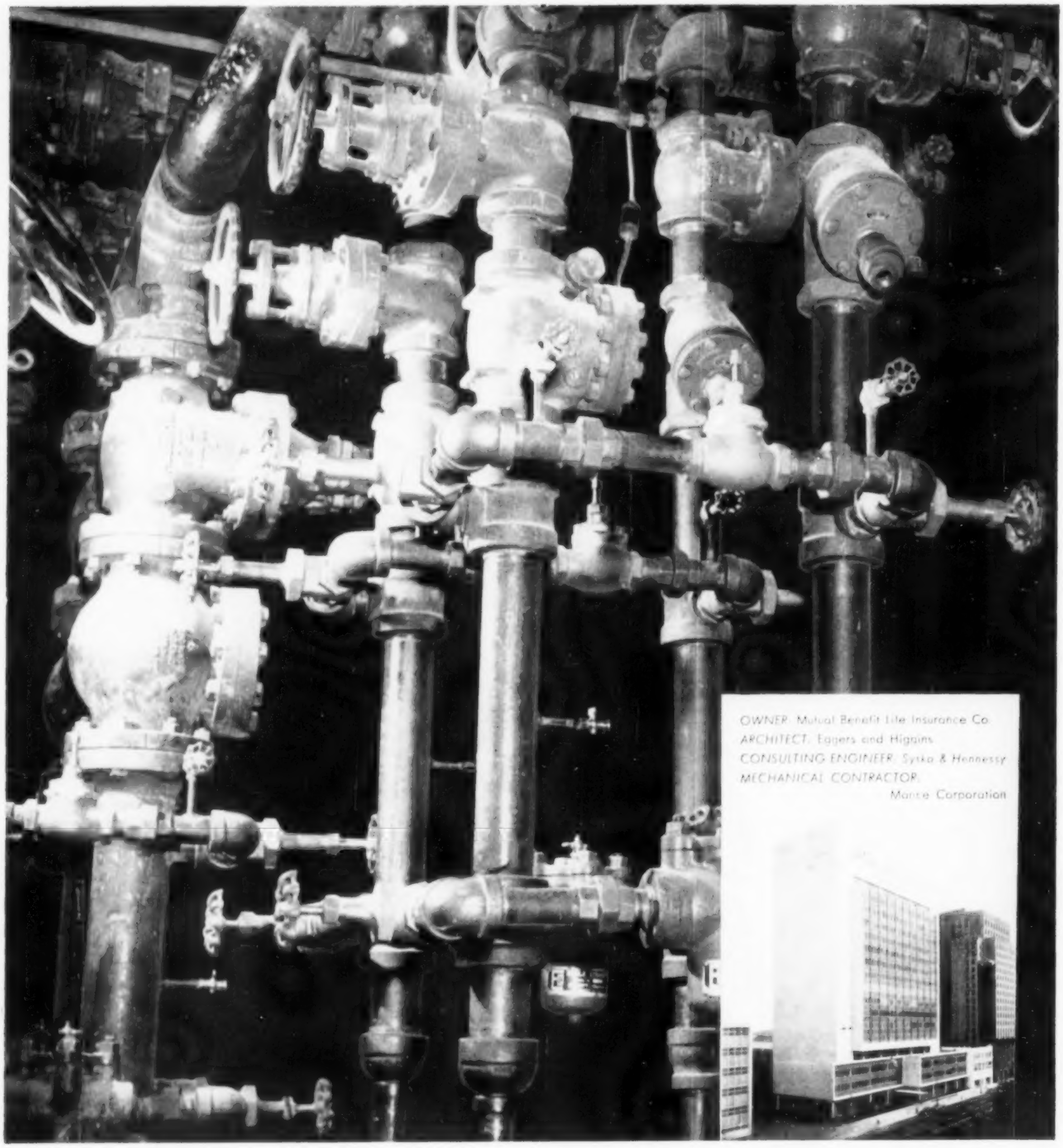
These features have won immediate approval for Yarway Welbond Valves in steam power plants everywhere. Now available in 9 sizes, ¼" to 2½", angle and straightway. Pressures to 2500 psi, temperatures to 1050°F. Write for Bulletin B-453.

YARNALL-WARING COMPANY
100 Mermaid Avenue
Philadelphia 18, Pa.
Branch Offices in Principal Cities



YARWAY WELBOND ... *is the valve for high pressures/high temperatures*

REPRESENTATIVE USERS OF YARWAY WELBOND VALVE: Houston Lighting & Power Co. Jersey Central Power & Light Co. Boston Edison Co. Virginia Electric & Power Co. Connecticut Light & Power Co. City of Independence, Mo. Aluminum Co. of America City of Pasadena, Calif. Mississippi Power & Light Co. Rochester Gas & Electric Corp. Ohio Edison Co. Central Hudson Gas & Electric Corp.



In its new 20-story Home Office Building in Newark, New Jersey,

THE MUTUAL BENEFIT LIFE INSURANCE COMPANY

USES **WALWORTH VALVES...100%**

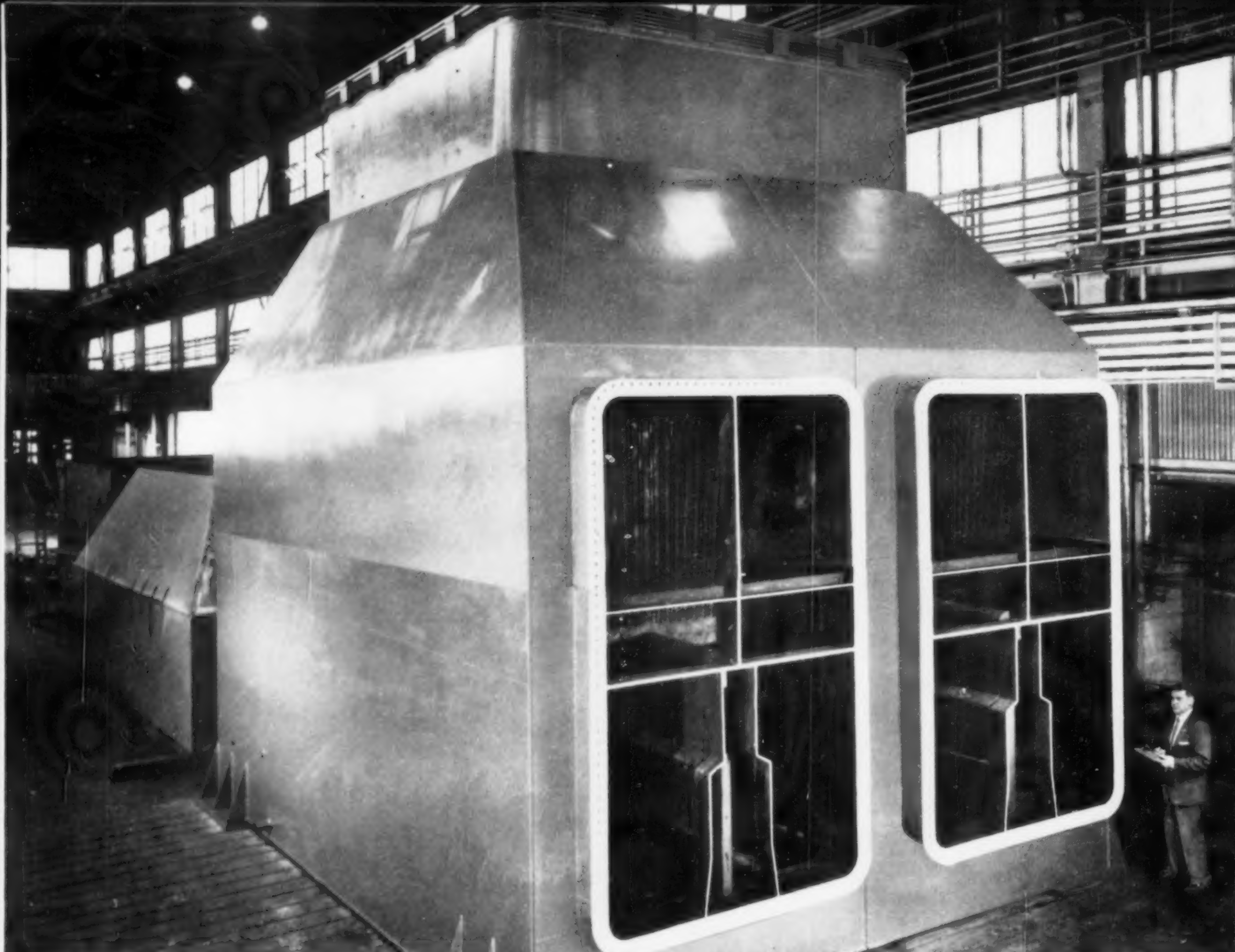
in both Plumbing and Heating Systems

WALWORTH

750 THIRD AVENUE, NEW YORK 17, N. Y.

DISTRIBUTORS IN PRINCIPAL CENTERS THROUGHOUT THE WORLD

WALWORTH SUBSIDIARIES: ALLOY STEEL PRODUCTS CO. • CONOFLOW CORPORATION • GROVE VALVE AND REGULATOR CO. M&H VALVE & FITTINGS CO. • SOUTHWEST FABRICATING & WELDING CO., INC. • WALWORTH COMPANY OF CANADA, LTD.



This 75,000 square foot condenser pre-assembled and pre-fitted at Maryland was shipped in 5 compact maximum rail clearance sections.

5-piece Maryland condenser design

Horizontal division separates the neck section (1). The main body of the condenser is divided both vertically and horizontally, as shown by sections numbered (2), (3), (4) and (5).

Maryland's 5-Piece Condenser

Designed for Low Cost Field Erection

Maryland designed, built, pre-fit and pre-assembled this 75,000 square foot condenser in five compact sections for accurate and low cost field erection.

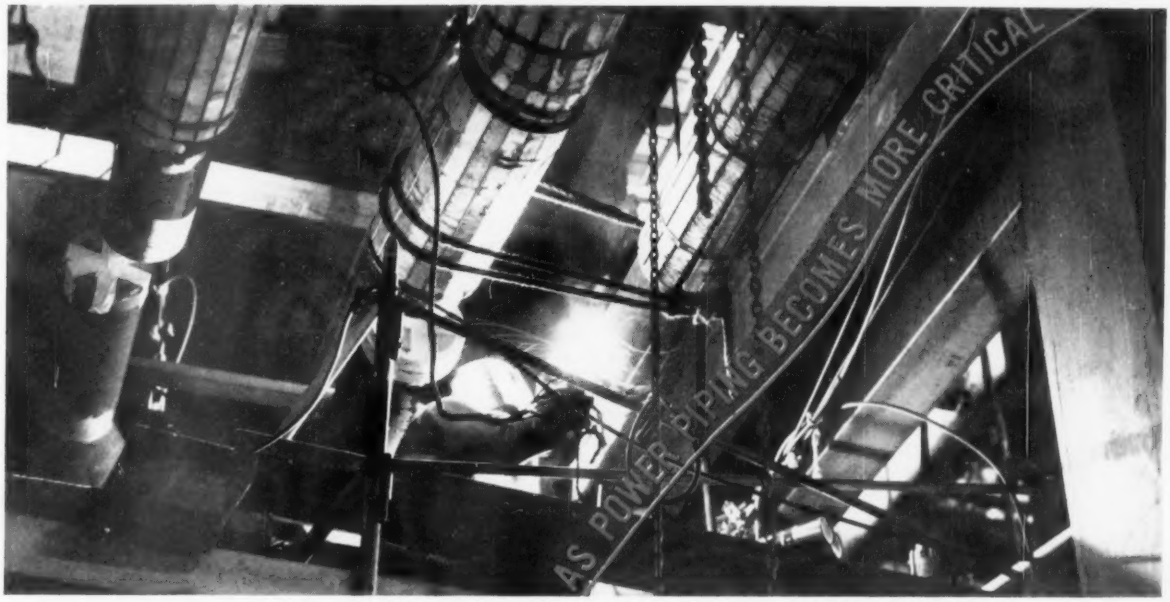
Now being installed at Reesedale, Pennsylvania in West Penn Power's Armstrong Station, this unit indicates Maryland's ability to engineer and build large surface condensers for electric utilities. It will condense 812,819 pounds of steam per hour exhausted from a 156,250 KW Westinghouse turbo-generator, and its full deaerating hotwell will produce condensate with no more than .01 cc of oxygen per liter.

When *your* plans call for steam surface condensers, consider the advantages offered by Maryland's years of engineering experience and facilities for designing and manufacturing condensers, of any size.

Industrial Products Division

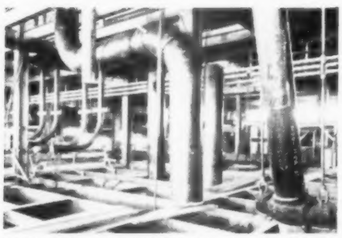
MARYLAND SHIPBUILDING & DRYDOCK COMPANY

BALTIMORE 3, MARYLAND • *Representatives in Principal Cities*

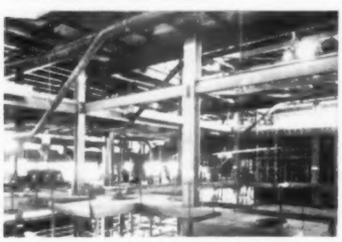


A main steam line is K-Welded to the mixing header. There are eight main steam lines to this header from the boiler, and four from mixing header to turbine. With inside diameters up to 4.938 in. and minimum wall thicknesses as much as 2.860 in., sections weigh four to five tons. Main steam header weighs over ten tons.

KELLOGG'S FIELD WELDING KEEPS PACE



Section of second cold reheat system (center) in process of being positioned, with section of first cold reheat system (right) partially in place. Four boiler feed risers at left.

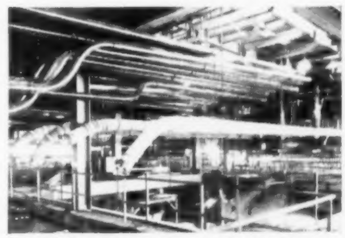


General view inside Eddystone Station, eleven floors high, showing high pressure boiler feed water risers (top and center) as these were being erected by Kellogg.

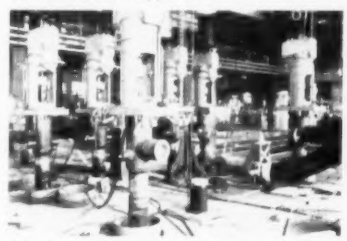
During erection of Philadelphia Electric Company's Eddystone Station, Unit No. 1, *eighty* K-Welds* . . . Kellogg's inert gas shielded technique of manual arc welding . . . will be made by Kellogg on the Type 316 stainless main steam lines alone. Following Kellogg's shop fabrication procedures on this job, every K-Weld will be given a total of *four* inspections by liquid penetrant and radiography during welding and after heat treatment.

Time required to finish and inspect each weld completely on this supercritical, heavy-walled piping is *eight to ten working days*. Further evidence of the magnitude and complexity of Kellogg's erection and welding assignment is provided by these construction photographs.

Kellogg welcomes inquiries on its complete service to the power piping industry from consulting engineers, engineers of power generating companies, and manufacturers of boilers, turbines, and allied equipment.



Four of the main steam lines, showing the protective wood lagging installed at Kellogg plant. This is not removed, except at welding points, until insulation is applied.



Six of the eight special high pressure valves being installed by Kellogg at the Eddystone Station—Philadelphia Electric Company's new supercritical steam power plant.

Fabricated Products Sales Division, THE M. W. KELLOGG COMPANY, 711 Third Avenue, New York 17, N. Y.

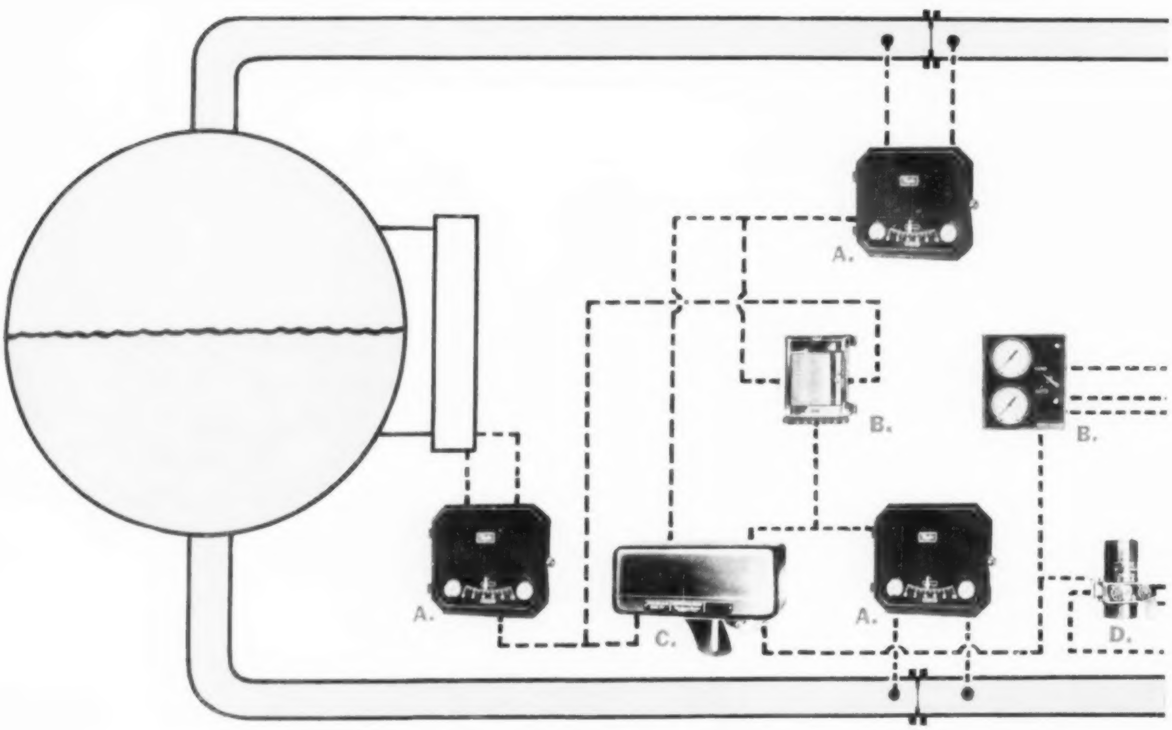
A SUBSIDIARY OF PULLMAN INCORPORATED

The Canadian Kellogg Co., Ltd., Toronto • Kellogg International Corp., London • Kellogg Pan American Corp., New York
Société Kellogg, Paris • Companhia Kellogg Brasileira, Rio de Janeiro • Compania Kellogg de Venezuela, Caracas

M W KELLOGG

POWER PIPING—THE VITAL LINK

** Registered trademark of The M. W. Kellogg Company*



Copes Feedwater Control brings stability to high-duty steam generators

Today's modern boiler design with higher ratings and fast load changes requires increased accuracy and more sensitive response in regulating water feed into the steaming boiler.

To meet these rigid demands, Copes Type 3-L Feedwater Control combines the dynamic balance of custom-engineered control valves with the positive performance of precision-built instruments.

Feed to the boiler is modulated by three influences—steam flow, feedwater flow and drum water level. Feedwater input closely matches steam output on constant loads or during periods of wide load changes. A stabilized water level is always maintained regardless of changes in load or feed pressure, and while blowing down or blowing soot.

Space-saving instruments record steam flow, feedwater flow and water level on linear strip charts. These charts give a 30-day continuous record, with a 3-hour record clearly visible. Standard-size recorders are also available.

A complete line . . . a complete service

Over 50 years of design experience backs Copes-Vulcan's broad line of control systems for boiler cleaning, combustion, feedwater, pressure reducing and desuperheating operations.

Bulletin 1038 shows feedwater control at Seward Station

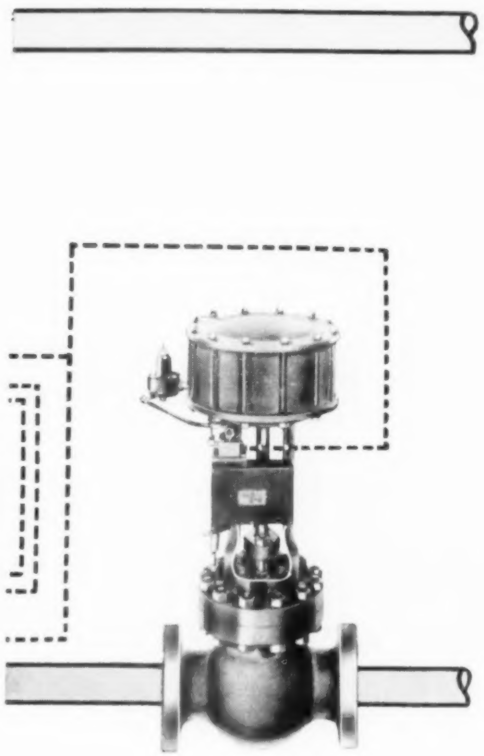
This twelve-page bulletin details Seward's complete control system. Schematic drawings show air flow and fuel loading loops. Boiler drawings locate Vulcan soot blowers and wall deslaggers. Write for your copy.

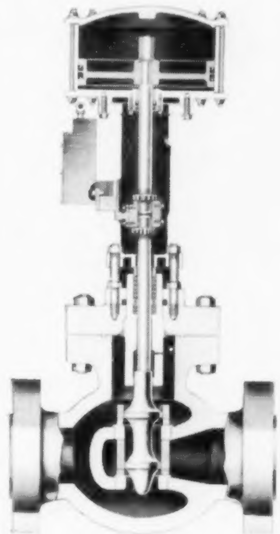
Copes-Vulcan Division

BLAW-KNOX COMPANY

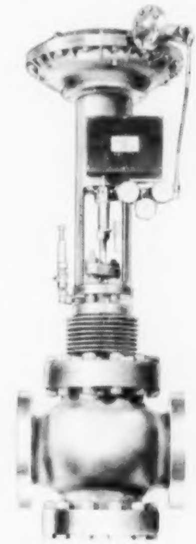
Erie, Pennsylvania

C-V NEWS NOTES





Type CV-P Valve



Type CV-D Valve

Maximum power with precise positioning is assured with Copes-Vulcan Type CV-P Valve. Responsive to most minute changes in actuating impulse, this rugged valve delivers top power at the valve stem. For less demanding service conditions, Copes-Vulcan builds a versatile diaphragm-operated Type CV-D Valve.

A. Indicating transmitters send water level and flow influences to the computing relay where steam-flow and water-flow are accurately balanced and water-level influence is relayed by air impulse.

B. Miniature recorders and transfer panel are ideal for console-type or graphic panels or for compact grouping on existing panels. Conventional recorders of standard size are also available.

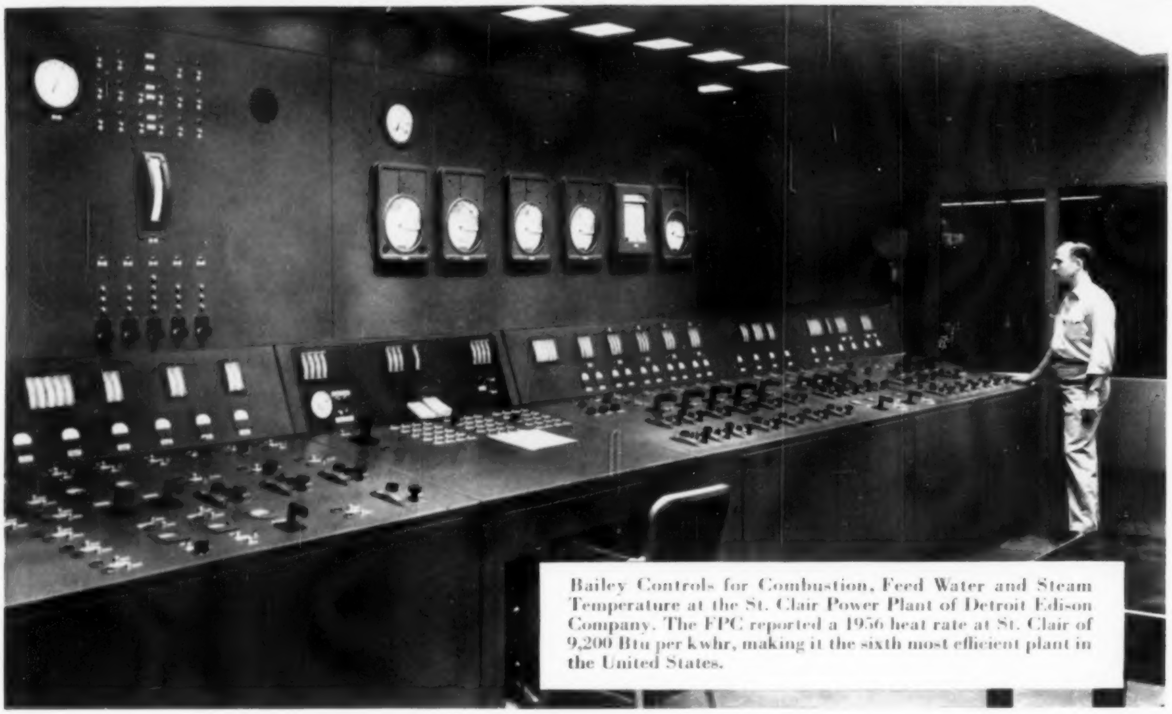
C. Compact beam-balanced type computing relay sends an air impulse to the controller and auto-manual transfer panel, either of which can closely control the feedwater valve position.

D. Transet controller has been designed to provide two-knob turning. Controller, transmitters and relays may be field mounted wherever most convenient without respect to location to each other.

Bevel-seated cylinder type
Rectangular ports Characterized V ports

Stream-flow type
Bevel-seated

The right fitting for the job. Copes-Vulcan valves are furnished for all capacities and pressures with closely balanced bevel-seated plunger and seats of stainless steel with stellite facing when required. Ports are job-engineered according to specified flow and pressure conditions. Wear on moving parts is minimized.



Bailey Controls for Combustion, Feed Water and Steam Temperature at the St. Clair Power Plant of Detroit Edison Company. The FPC reported a 1956 heat rate at St. Clair of 9,200 Btu per kwhr, making it the sixth most efficient plant in the United States.

Why Bailey is the choice of America's most efficient* STEAM PLANTS!

Take the top six on the Federal Power Commission's heat rate report. All use Bailey Meters and Controls. Five of the six chose Bailey *exclusively!*

Coincidence? Let's take a *larger* sample. Out of the top 46 "most efficient" power plants listed, 38 use Bailey products. Here's why:

1. A Complete Line of Equipment

You can be sure a Bailey Engineer will offer the right combination of equipment to fit your needs. Bailey manufactures a complete line of standard, compatible pneumatic and electric metering and control equipment that has proved itself. Thousands of successful installations involving problems in measurement, combustion, and automatic control are your assurance of the best possible system.

2. Experience

Bailey Engineers have been making steam plants work efficiently for more than forty years. Veteran engineer and new engineer alike, the men who represent Bailey, are storehouses of knowledge on measurement and control. They are up-to-the-minute on the latest developments that can be applied to your problem.

3. Sales and Service Convenient to You

There's a Bailey District Office or Resident Engineer close to you. Check your phone book for expert engineering counsel on your steam plant control problems.

*6 MOST EFFICIENT PLANTS—1956 Heat Rates Reported by FPC

	Btu/kwhr	Meters	Combustion Control	Feed Water Control	Supht. Control
1. Tanners Creek (Indiana and Michigan Electric Company)	9,106	B	B	B	B
2. Kanawha River (Appalachian Electric Power Company)	9,115	B	B	B	B
3. Kyger Creek (The Ohio Power Company)	9,176	B		B	B
4. Muskingum River (The Ohio Power Company)	9,176	B	B	B	B
5. Clifty Creek (Indiana-Kentucky Electric Company)	9,200	B	B	B	B
6. St. Clair (The Detroit Edison Company)	9,200	B	B	B	B

A140-1

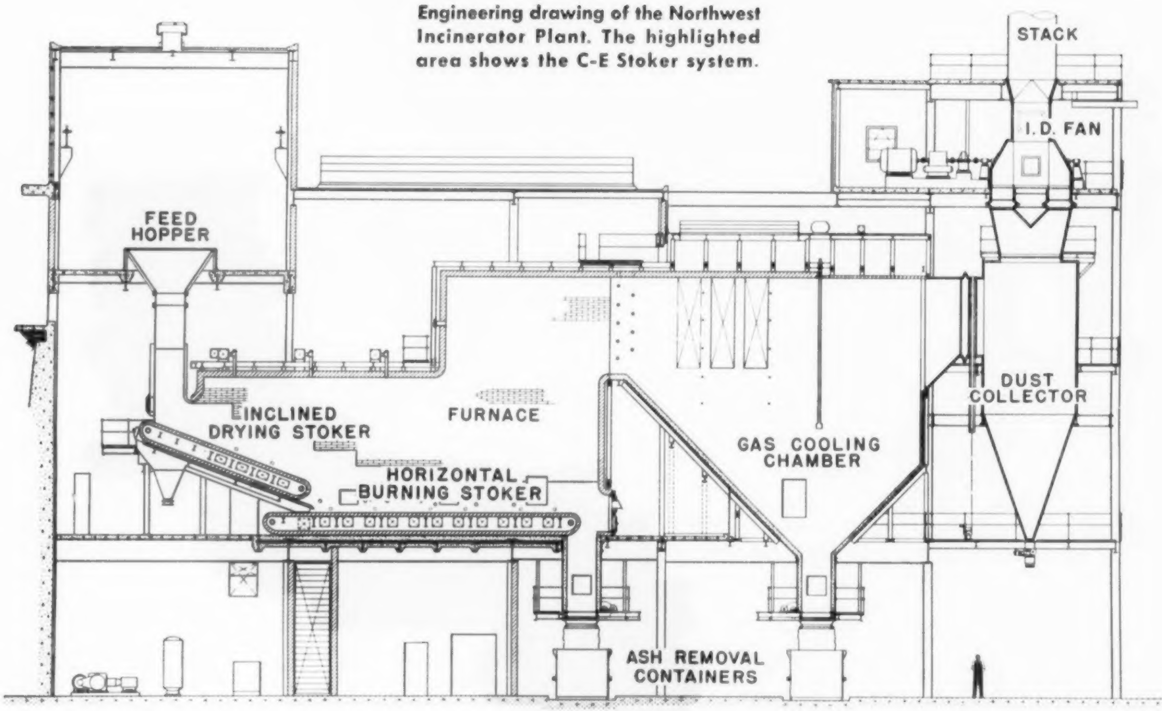
BAILEY METERS

Instruments and controls for power and process

BAILEY METER COMPANY

1025 IVANHOE ROAD • CLEVELAND 10, OHIO

In Canada—Bailey Meter Company Limited, Montreal

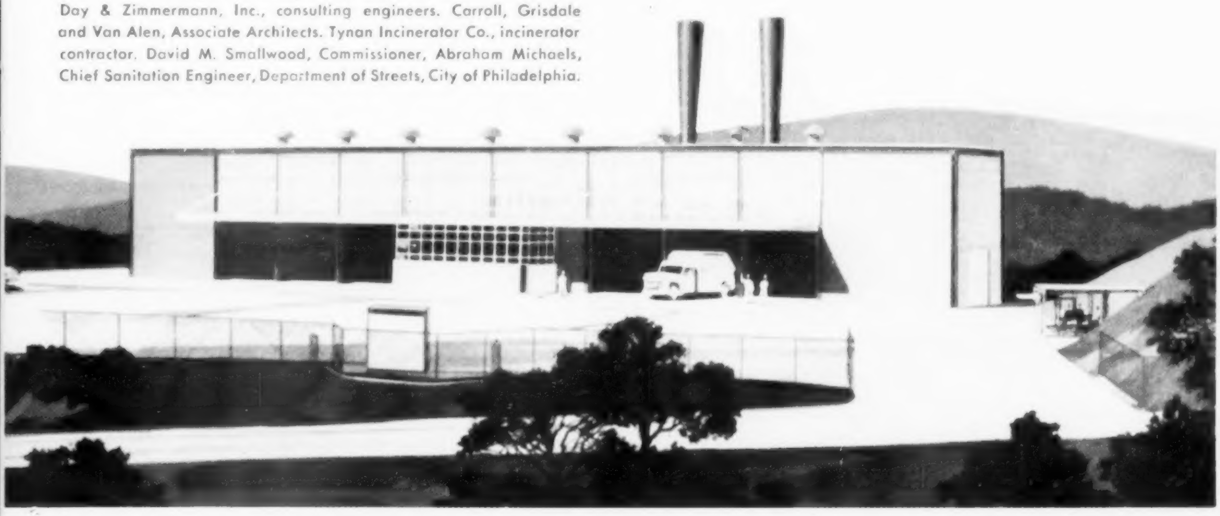


Engineering drawing of the Northwest Incinerator Plant. The highlighted area shows the C-E Stoker system.

HIGHLIGHTS OF C-E DESIGN

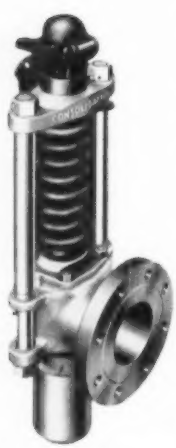
- Large hopper and chute provide sealed, continuous supply of refuse.
- Feed of refuse is unobstructed, non-clogging.
- Advance of fuel bed from front to rear is positive, continuous.
- Supply of air to fuel-bed is zone-controlled.
- Residue is discharged continuously from grate surface.
- Operation is easy — no heavy stoking or cleaning of fires.
- Chrome cast iron — most suitable material — is used for grate surface.
- Wide keys over driving chains prevent fouling.
- Grate elements are easily replaceable.
- Steel driving chains take all tension.
- Take-up mechanism is easily accessible at front.
- Maximum availability is assured.
- Stokers are applicable to largest incinerators.
- Less labor is required than with any other burning method.

Architect's drawing of Philadelphia's Northwest Sanitation Facilities. Day & Zimmermann, Inc., consulting engineers. Carroll, Grisdale and Van Alen, Associate Architects. Tynan Incinerator Co., incinerator contractor. David M. Smallwood, Commissioner, Abraham Michaels, Chief Sanitation Engineer, Department of Streets, City of Philadelphia.



STOPS VALVE LEAKAGE

at high pressures



Consolidated "Maxiflow" Safety Valve, Type 1700 Series, Sizes: 1½" thru 6".

The Thermodisc Seat in Consolidated "Maxiflow" Safety Valves solves the problem of valve leakage at high pressures, saves steam and reduces maintenance.

Research has shown that when a safety valve on high pressure service reseats after popping, a small amount of leakage occurs momentarily between seat and disc before the valve seals off and becomes tight. This leakage is comparable to steam flow through small, individual orifices.

When a small steam leak exists at some point on the valve seat, the area near the point is cooled by the refrigeration effect of the escaping vapor. The metal tends to contract causing slight deformation of the seat. This acts to increase the gap size between disc and seat bushing at the leakage point. The size of the gap increases rapidly until the rate of leakage becomes extremely high.

The Thermodisc Seat prevents seat distortion and consequent leakage with a seat element recessed to form a thin wall at the area of seat contact. This design permits a high rate of heat-transfer in the metal *that eliminates all thermal differences.*

The result is a degree of tightness, essential to the operation of safety valves in high pressure services, never previously achieved. For complete details, write for Bulletin 707B.

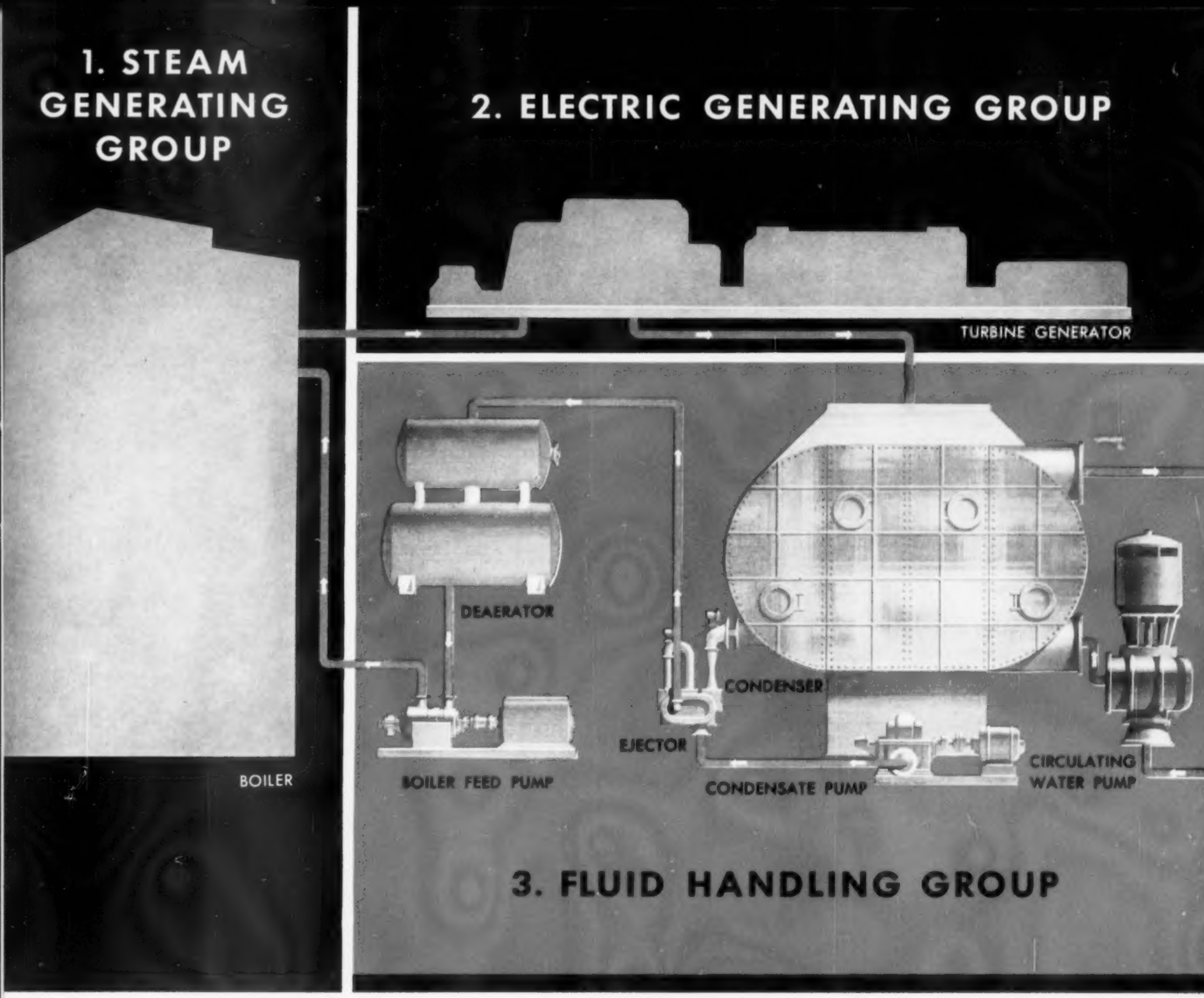
MAXWELL — M&M — MANNING — & MOORE INC. — TRADE MARK

CONSOLIDATED SAFETY VALVES

A product of

MANNING, MAXWELL & MOORE, INC.

Consolidated Ashcroft Hancock Division • Stratford, Connecticut
In Canada: Manning, Maxwell & Moore of Canada, Ltd., Galt, Ontario

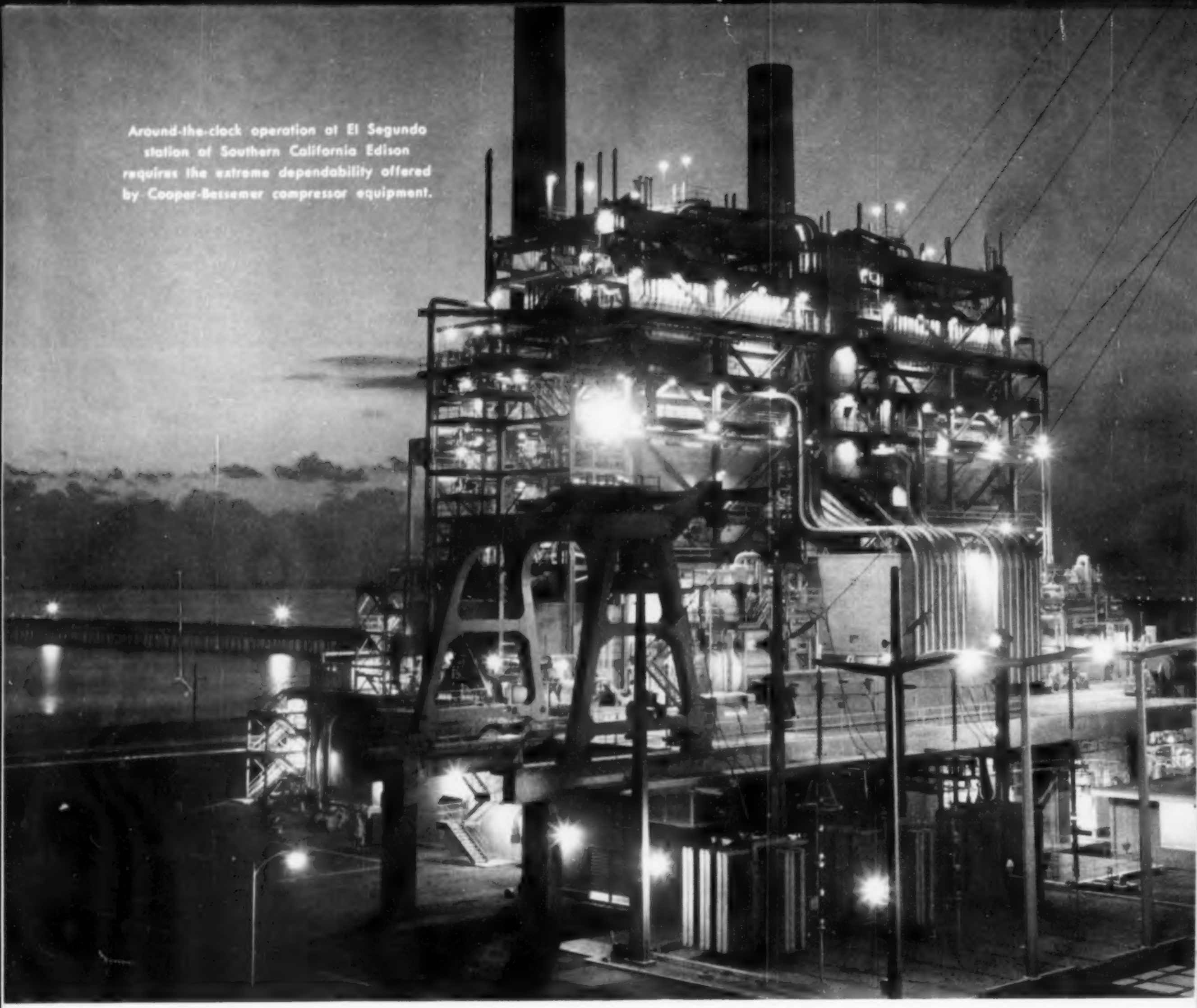


LOOK AT ALL THREE FOR POWER

Power plant reliability often depends on the effect of one component of the *Fluid Handling Group* on another. As the manufacturer of all major components of this group, Worthington has a reservoir of experience and knowledge that can benefit you. This "system-wise experience" with modern complex plant equipment can help solve your fluid handling problems. To put "system-wise experience" to work get in touch with your nearest district office. Worthington Corporation, Harrison, New Jersey.

45-3

WORTHINGTON



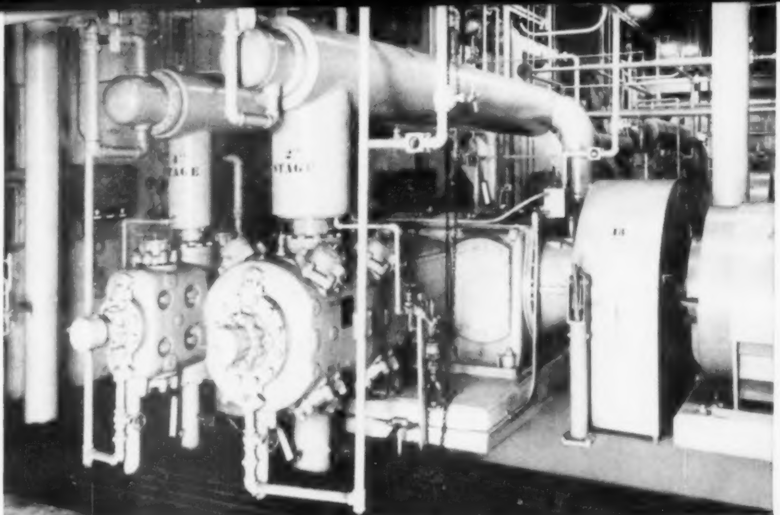
Around-the-clock operation at El Segundo station of Southern California Edison requires the extreme dependability offered by Cooper-Bessemer compressor equipment.

Increase boiler capacity,

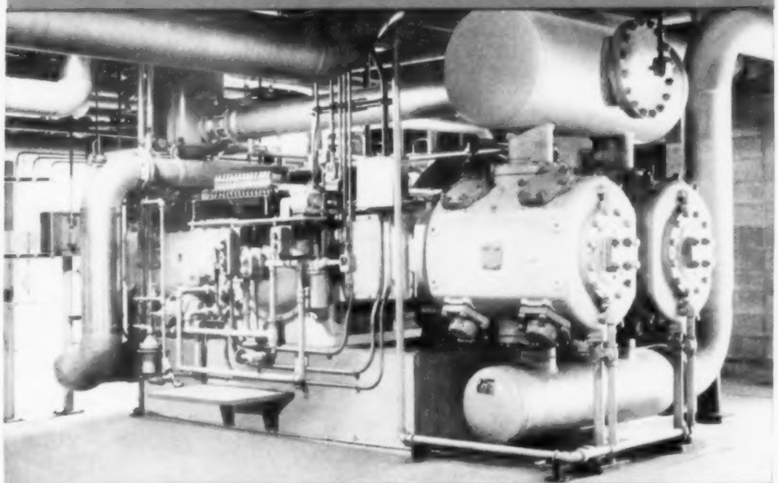
...with Cooper-Bessemer air Compressors for soot blowing service

Steam stations everywhere profit from multi-stage Cooper-Bessemer M-Line air compressors used to clean slag and ash deposits from boiler unit surfaces. For example, four steam stations of Southern California Edison Company, the Sayreville station of Jersey Central Power & Light Company, the Breed Plant No. 1 of Indiana and Michigan Electric Company, and the Philip Sporn Plant Unit No. 5 of the Ohio Power Company . . . *all* depend on the experience and efficiency offered by Cooper-Bessemer.

You'll find air cleaning with Cooper-Bessemer equipment the most effective and economical answer to a continuing problem. Automatic regulation, heavy-duty con-

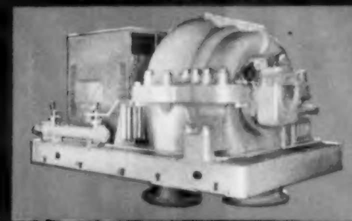


Extreme accessibility is another plus feature of motor-driven FM compressors in the Sayreville station of Jersey Central Power & Light.

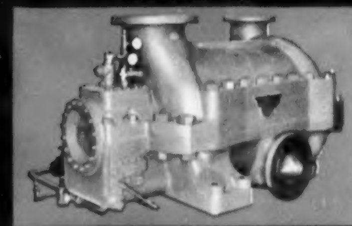


This compact Cooper-Bessemer FM compressor, rated 500 hp at 400 rpm, cuts trouble and costs at the El Segundo steam-electric generating station.

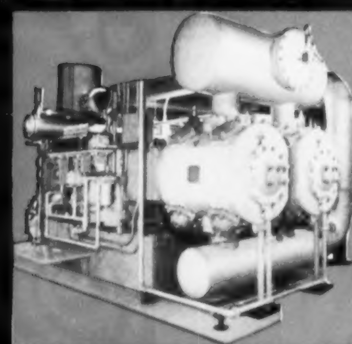
TO MEET SEVERE SERVICE REQUIREMENTS



Direct motor-driven multi-stage centrifugal compressors for sizes from 3000 cfm free air and up.



Multi-stage centrifugal compressors for sizes from 7000 cfm free air and up.



Multi-stage reciprocating compressors for up to 30,000 cfm free air.

availability and efficiency

struction and a range of horsepower sizes offer you the ideal unit.

For your next steam generating unit, check with Cooper-Bessemer for the latest in soot blowing services. Sizes are available up to 10,000 hp in both reciprocating and centrifugal designs. Write for additional information.

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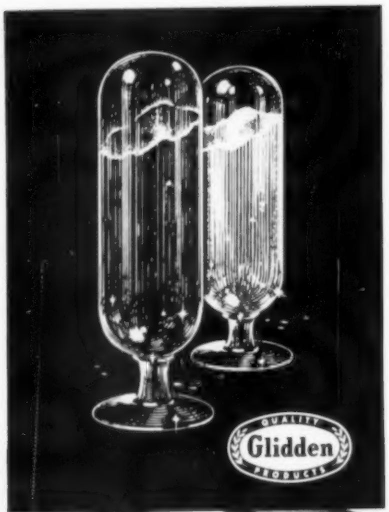
Cooper Bessemer

GENERAL OFFICES: MOUNT VERNON, OHIO

ENGINES: GAS · DIESEL · GAS-DIESEL
COMPRESSORS: RECIPROCATING AND CENTRIFUGAL
ENGINE OR MOTOR DRIVEN



Power for producing Republic

Glidden pigments starts with
ELECTRUNITE Boiler Tubing

Keeping pigment production up is a big job that demands power plant dependability. And dependable power starts with Republic ELECTRUNITE® Boiler Tubing at the new Ardian Joyce Works of the Chemicals-Pigments-Metals Division, The Glidden Company, Baltimore, Maryland.

Power for this pigment production comes from units of 70,000 pounds per hour with 725 p.s.i. saturation temperature. These units were designed, constructed, and installed by Riley Stoker Corporation. Republic ELECTRUNITE was used throughout for the boiler waterwalls and the economizer.

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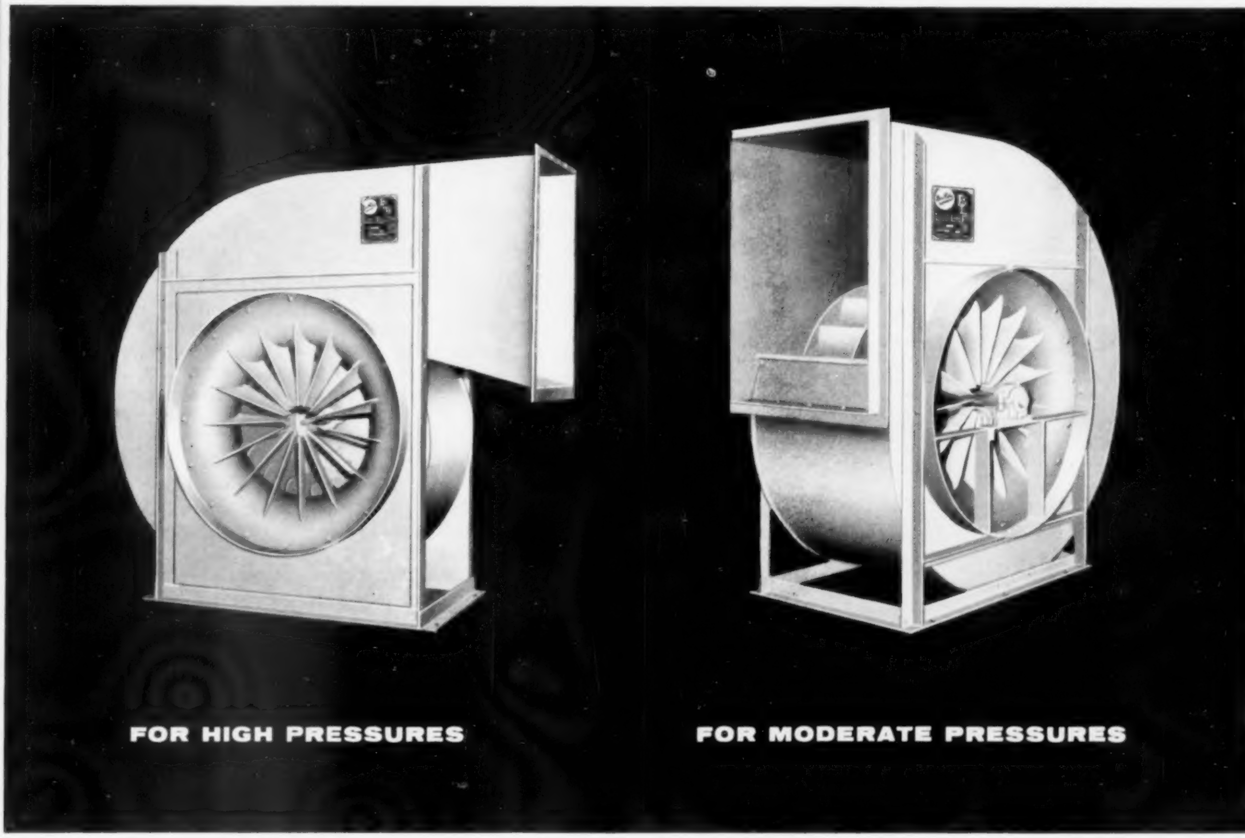

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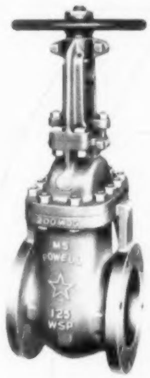

Fig. 1793—Large Iron Body Bronze Mounted Gate Valve for 125 pounds W.P. Outside screw rising stem and yoke. Solid wedge disc. Valves up to 8" can be supplied with split wedge disc.

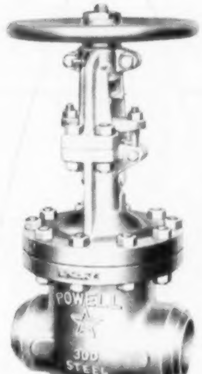

Fig. 3003 WE Steel Gate Valve for 300 pounds. Outside screw rising stem and yoke, bolted flanged bonnet. Powell steel valves are available for pressure from 150 through 2500 pounds.

Fig. 11313 Steel 1500-pound Pressure Seal Gate Valve with By-pass. Sizes, 4" to 14", inclusive. By-pass valve is Fig. 1333-A 1500-pound Integral Bonnet Valve.

Fig. 1333-A Small Steel Integral Bonnet Angle Valve for 1500 pounds. One-piece construction, compact design. Socket weld ends. Sizes ½" to 2". 2500 pound valves also available in sizes ½" to 3".

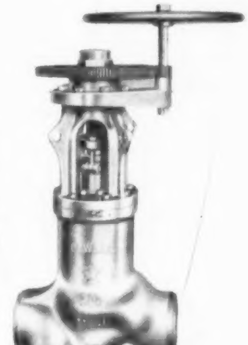

Fig. 19084 Steel Pressure Seal Globe Non-return Valve for 900 pounds. Spur gear operated. Can be furnished with bevel gears.

Fig. 375 Bronze Gate Valve for 200 pounds W.P. Union bonnet, inside screw rising stem. Renewable accurately guided nickel bronze solid or split wedge discs.

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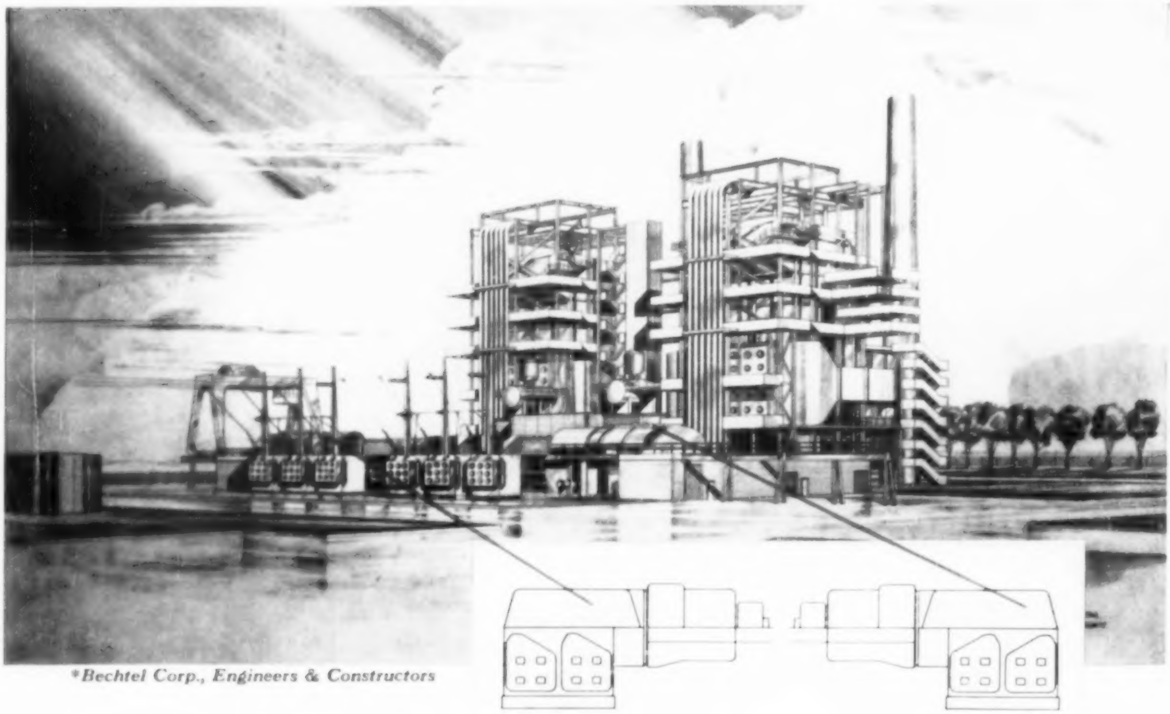

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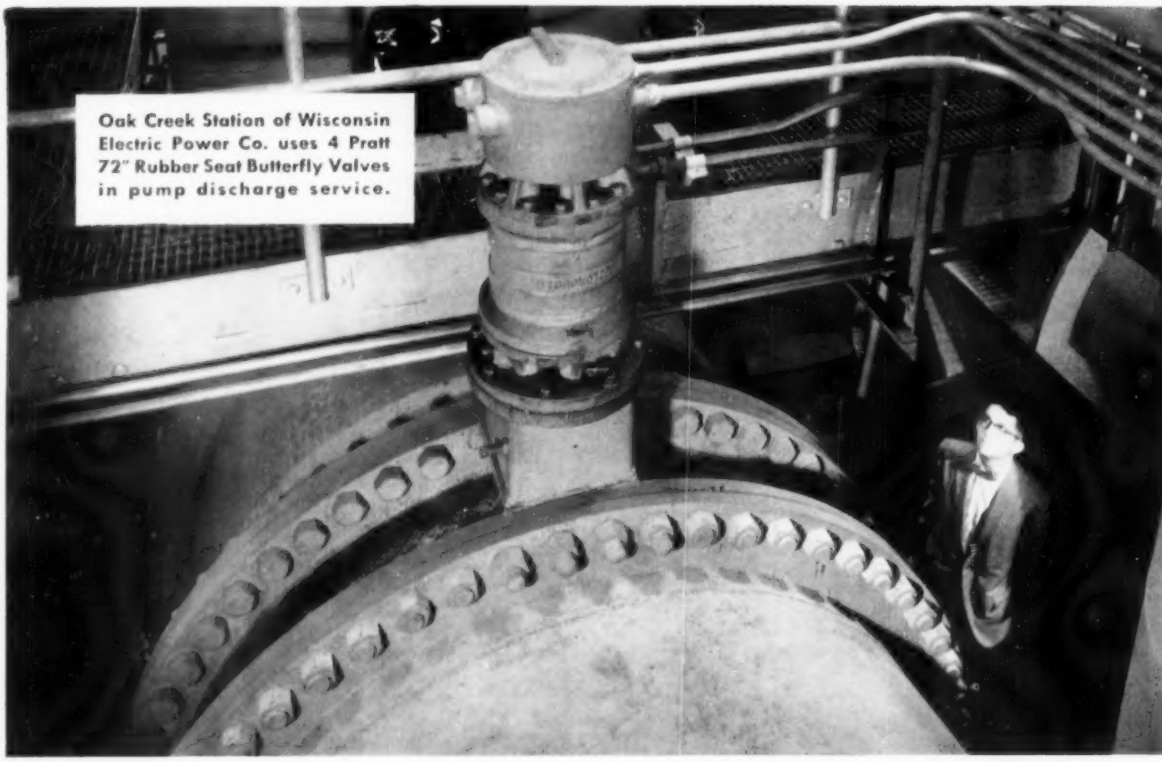

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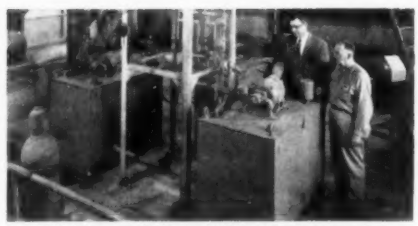
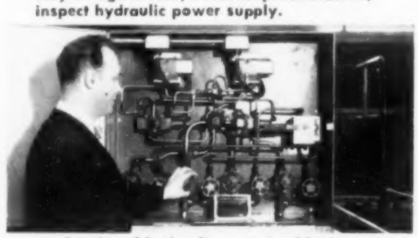


Felix Kazmerchak, Ass't. Plant Engineer, and Ray F. Egebrecht, Pratt Representative, inspect hydraulic power supply.



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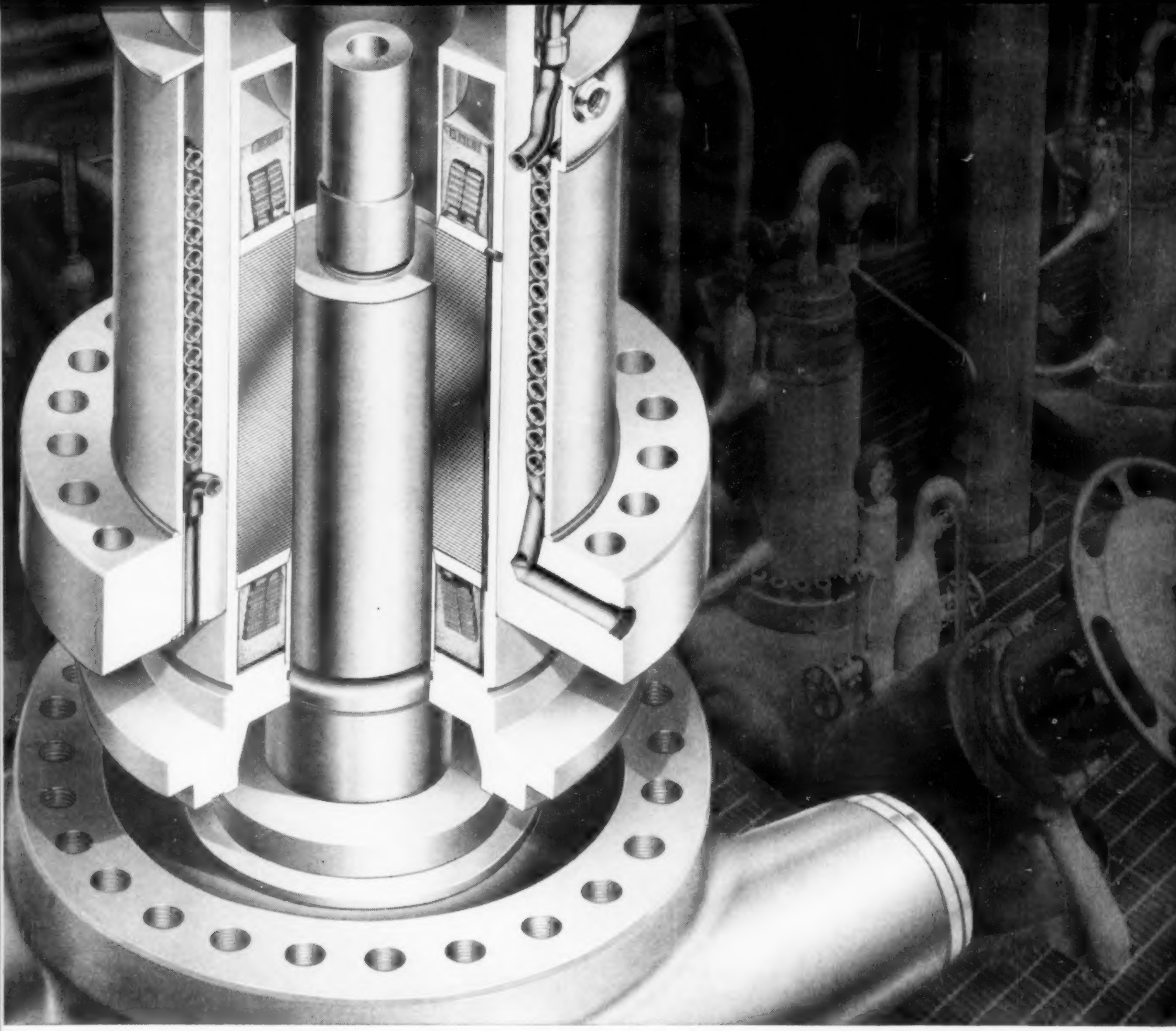

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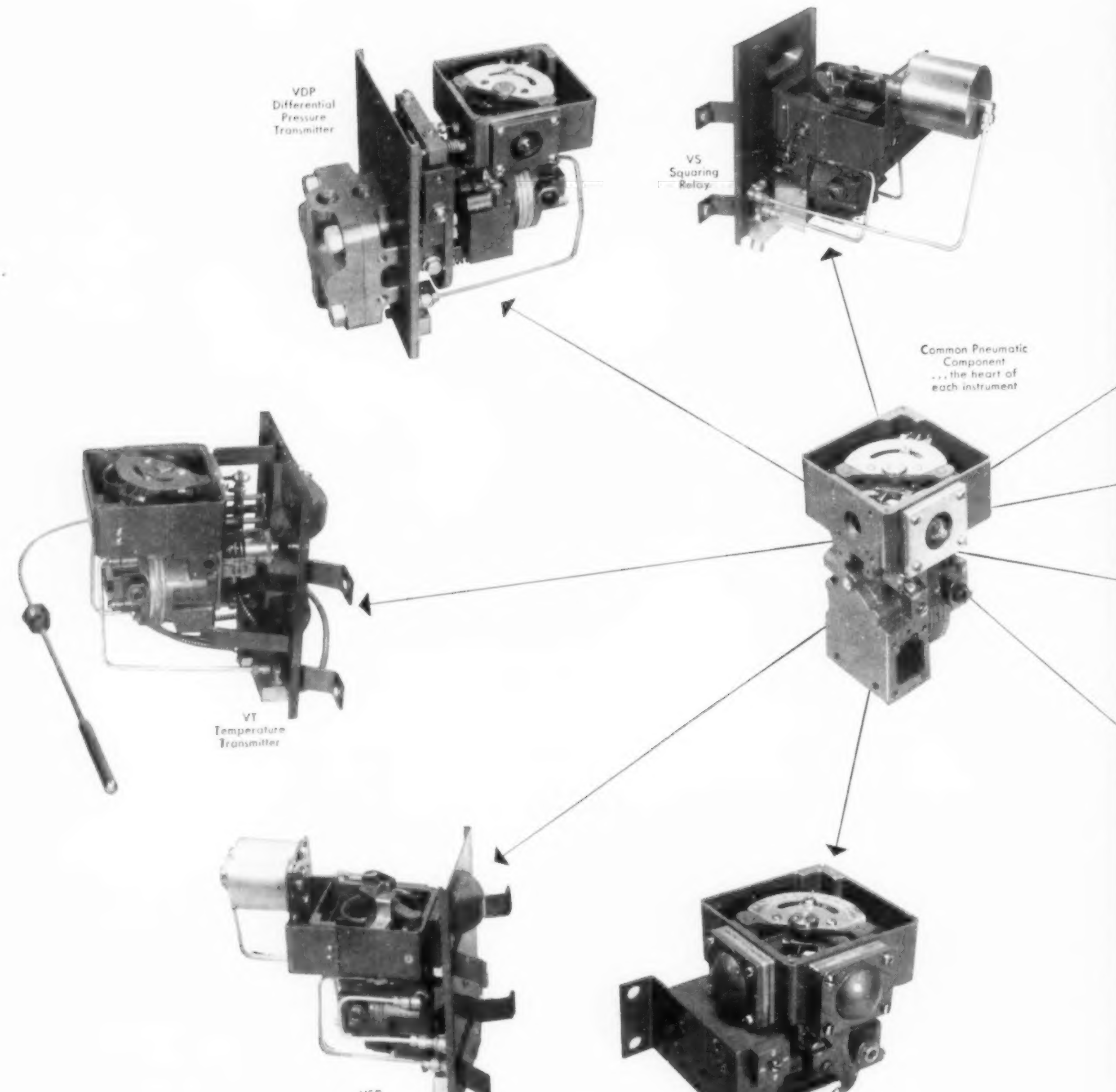

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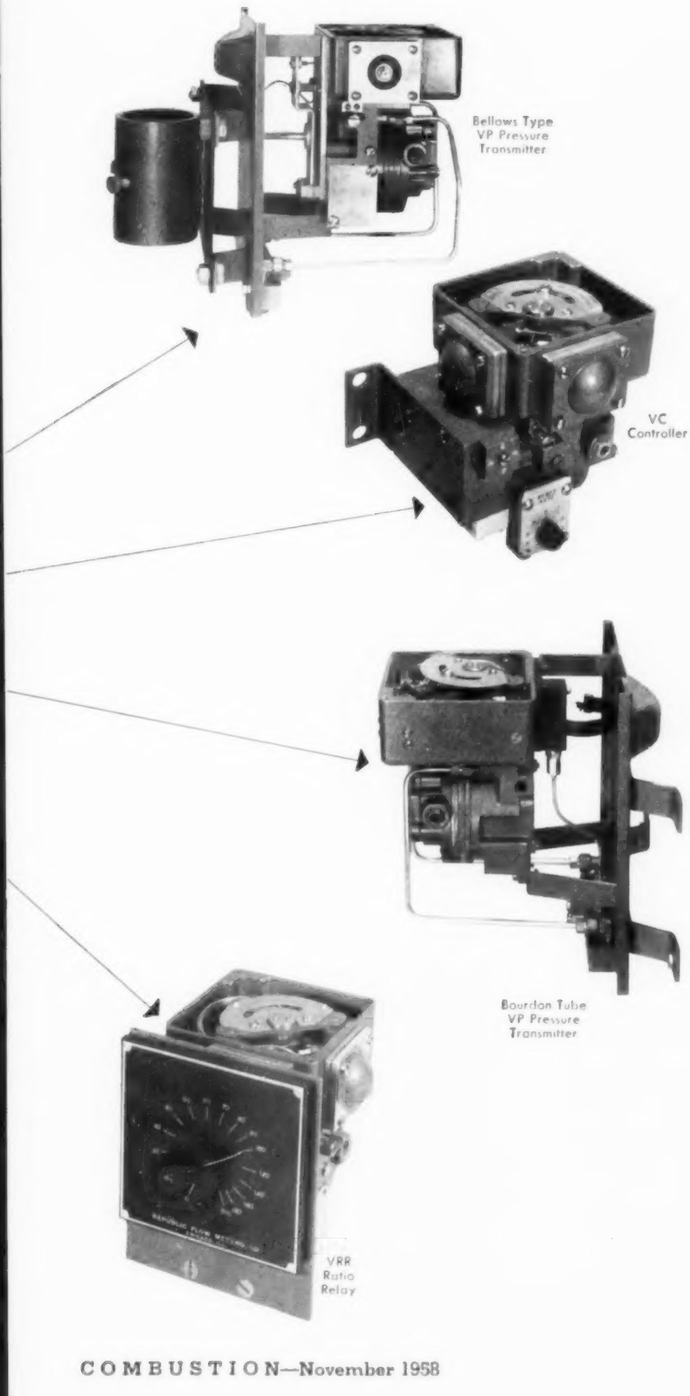

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Editorial

Dotting the "I's" and Crossing the "T's"

The recently concluded United Nations Second International Conference on the Peaceful Uses of Atomic Energy did not, from all reports we have seen, reveal any appreciable change in the economic future for nuclear power from that envisioned at the First Conference in 1955. We recall quite clearly the statement of Ragland, O'Neil and Habicht of American Blower Corp. in their paper before the American Power Conference of 1956 following that earlier Conference. We quote, "Despite the romantic lure of atomic considerations the real back-breaking job that lies immediately ahead is the building of this vast proposed outlay of conventional generating plants." Over two years and a recession later the job still seems to be one of developing more and better thermal plants.

Unless some new concept of electric generation can be devised the plants resulting from this need for new capacity will follow the expected evolutionary advances along the traditional lines of a live industry. Those advances will come about as much from attention to detail and checking and rechecking the existing scheme of things as from any other cause. Dotting the i's and crossing the t's still is a profitable way of life.

Such attention to detail and constant rechecking of the scheme of things produced two excellent papers to our way of thinking at the National Power Conference in Boston in September. Both papers were on the ever present boiler feed-pump and we plan to publish both. The one, "Pumping Power In the Feedwater Cycle" by Sam Arnow and J. L. Allen of Philadelphia Electric Co., is the lead article of this issue and the second "Evolution of the Boiler Feed-Pump Drive" by R. A. Baker, Public Service Electric and Gas Co., will appear next month. Both papers reflect the industry's deep concern with the awesome size of today's boiler feed-pumps and their proper application. Furthermore, both reveal the industry's willingness to check and recheck the role of the individual equipment implementing the power cycle.

This industry-wide willingness, or compulsion perhaps, to constantly examine the existing power plant in its every detail keeps the consultant, the designer and the supplier on their mettle. As trying as this way of life must be it offers, we feel, the only assurance that our power generation needs will be met as they present themselves and moreover that the need will be resolved with plants of reasonable yet progressive design.

Although the boiler feed pump may be called the heart of the power plant, it is not the whole thing, the authors state, and its position in the cycle must of necessity be governed by the larger consideration of plant pressure and temperature parameters, characteristic of load, cost of fuel, and so on. It will be the purpose of this paper to describe some actual pump installations and to discuss the conditions which led to the type of cycle adopted.

Pumping Power in the Feedwater Cycle*

By S. M. ARNOW and J. L. ALLEN

Philadelphia Electric Company

IT IS well known that less power is used to pump cold water than hot, so that placing the boiler feed pump ahead of the heaters results in a saving of pumping power; also the shaft sealing problem is considerably simplified. On the other hand the heaters operate at higher pressure which not only adds to the first cost of heaters, valves and piping, but subjects them to much more severe operating conditions.

In case of tube failures it is infinitely more costly to open a high-pressure heater with its complicated and heavy closure arrangement than to remove a few bolts which are sufficient to keep a low-pressure heater tight. Moreover, if the feed pump is placed after the last heater, the cycle gets the benefit of the temperature rise in the feedwater due to the work of the pump, which is by no means inconsiderable. In very high-pressure plants, with a desired maximum water temperature, this arrangement permits the highest pressure bleed to be dropped as much as 15–20 psi with no change in final feedwater temperature which in the high-pressure range may amount to a better utilization of work from the extraction point and result in a better thermal cycle, hence a considerable monetary saving.

Variable Speed Operation

As to variable speed, some power may be saved by resorting to a step device but one must consider the fact that unless a turbine drive can be incorporated in the cycle, the attainment of this saving inevitably leads to complication of the cycle by the need for additional equipment, such as speed increasers and hydraulic couplings, and the like, at a high first cost plus increased maintenance. The savings are marginal at best, and an occasional loss of the plant due to this extra equipment, may well wipe out the theoretical saving. In order to show the inherent limit of possible savings Fig. 1 shows a pumping system consisting of throttle pressure and friction which varies as the square of the flow; the possible saving in power is shown in the shaded area. In a boiler feed system, the greatest resistance is the boiler operating pressure which remains practically constant at all loads. The power savings therefore due to variable speed operation result from having to develop only sufficient power to satisfy the system requirements; i.e., flow and system head rather than the pump head characteristic (at constant speed) for the same flow demand. This, being a relatively minor component of the total system power demand, can result at best in only a modest saving. However, if the system resistance consists entirely (or mostly) of friction, a considerable saving would result. When one takes into consideration the linear drop in efficiency of the variable speed device, the net saving is small indeed. Moreover when the plant operates near full capacity, it has the constant drag of some five per cent attributable to coupling and or gear losses. All this makes artificial variable speed savings of dubious validity in a typical feedwater system, if power saving is the only consideration. The whole subject of such an installation should be approached with extreme caution. Many of our investigations showed that the possible return did not justify the added capital in-

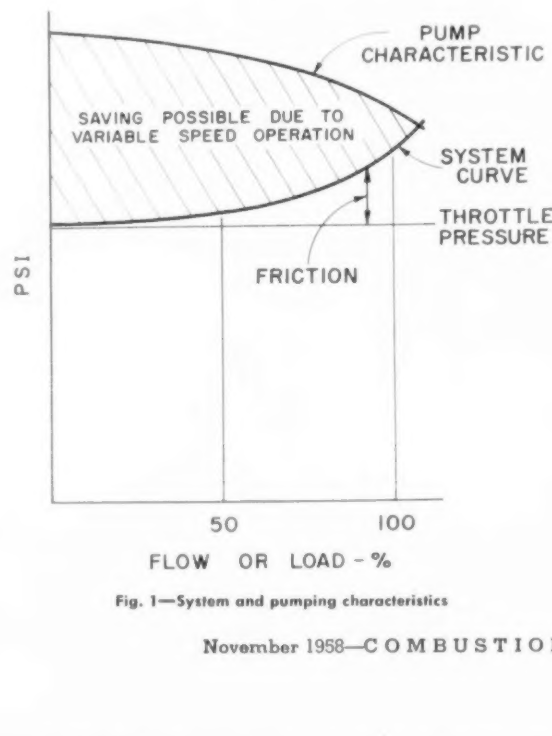


Fig. 1—System and pumping characteristics

* Contributed by the Power Division of the ASME for presentation at the ASME AIEE Power Conference, Boston, Mass., September 28–October 1, 1958 as Paper No. 58-PWR-2.

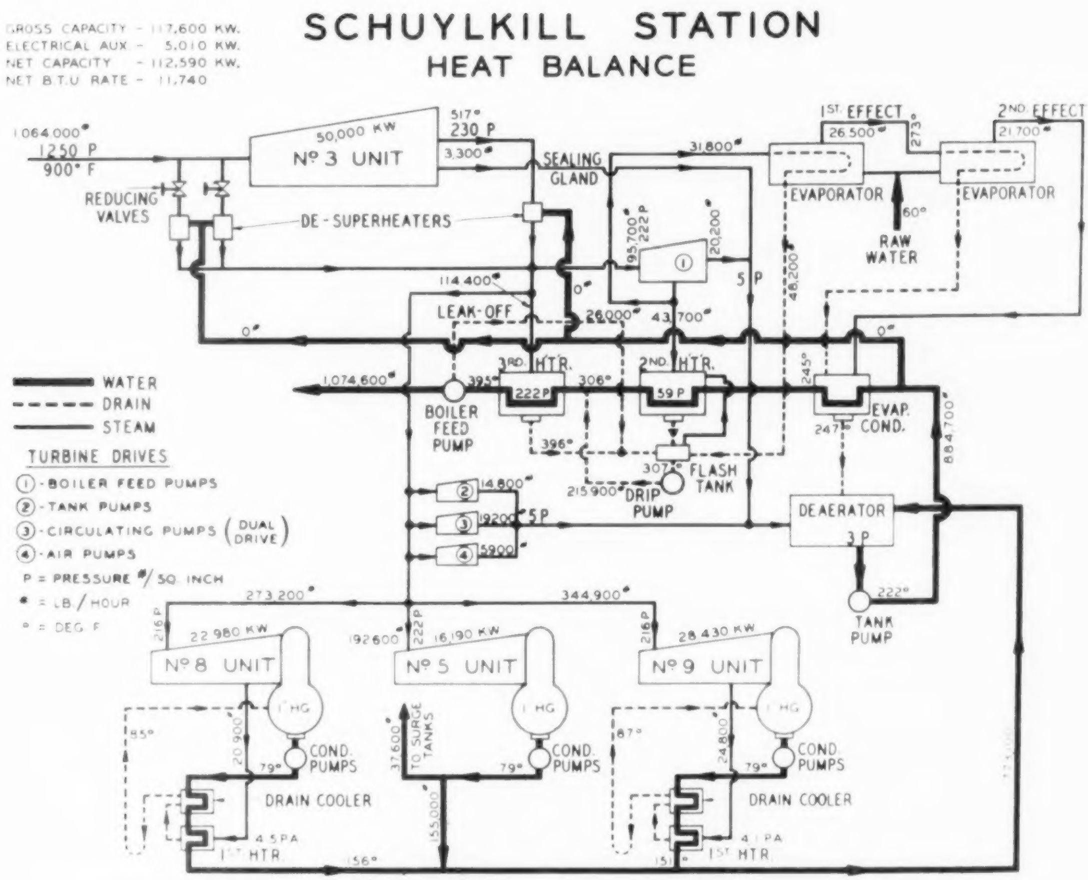


Fig. 2—Pumping arrangement for a superposed turbine

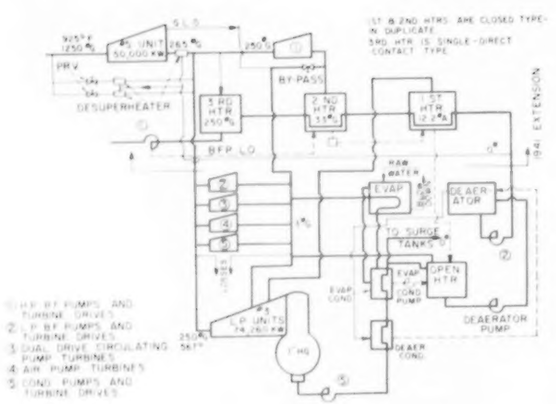
vestment. However as will be shown later, under certain circumstances the installation of variable speed devices may be justified or required.

Specific Cases

Many will remember the early slogan adopted by pump manufacturers when they attempted to replace reciprocating pumps with centrifugal units; namely, "you don't have to worry about a centrifugal pump working at no flow, it will merely churn the water." Well, perhaps one does not have to worry about that particular pump under such operation but it would be well to have a spare unit available. With modern high-efficiency pumps, the running clearances are very close in order to keep inner stage leakage to a minimum. Since the power requirement at no load, in such a pump, may well be 35 per cent of full load, the heat equivalent of this work will quickly cause the pump to seize, with the resultant long and costly repair job familiar to most power-plant designers and operators. In order to prevent such damage (due either to the reduction of load or the closure of a check valve), it becomes necessary to provide a means for bypassing a minimum flow of about 15 per cent through the pump at such times. Of course a bypass, if allowed to operate constantly, would entail a significant waste of pumping power, so one arranges for the bypass valve to open only if the load drops below a predetermined minimum. There are many ways to actuate a bypass device, the difficulty is what to do with the water after it has gone through the bypass. The water being handled (at an extremely high pressure and temperature) passes through the valve into a much lower pressure region. All the valve can do to reduce the water pressure is to convert it into high velocity. This energy must be dissipated before it can strike a section of pipe or fitting and bore right through it. Very careful and sober consideration must be given to this phase of the problem if one expects continuous trouble-free operation.

Case A

Fig. 2 shows the heat cycle and pumping arrangement of a superposed turbine installation. The old plant was 215-psia and 550 F. The topping pressure and temperature required to maintain a high-pressure turbine exhaust which would approximate the existing low-pressure plant steam conditions were 1250 psi and 900 F. The original turbines of 1915 vintage were tandem compound having a crossover pressure of 5 psia at full load. This was an ideal operating condition for the first bleed. There were a number of steam-driven auxiliaries exhausting to a heater at about 1-psi. Moreover all the dual-driven circulating pump turbines had ample steam capacity to provide adequate control of the exhaust system pressure at all loads. The top water temperature could be maintained at about 390 F



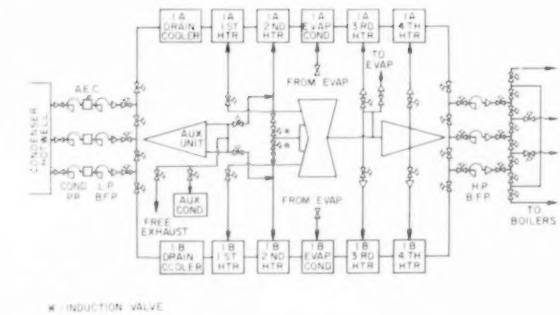


Fig. 4—Southwark station—heat cycle

Fig. 3—Chester station—heat cycle

by a heater utilizing the steam from the HP-turbine exhaust which was held constant at all loads. This meant a feed-heating step from 215 F to 390 F which was obviously not economical. The solution was to install high-pressure turbine-driven boiler feed pumps using their exhaust to obtain an intermediate feed-heating stage at say about 75-psia.

Detailed studies showed that this was by no means a simple task. At that time, the price of closed heaters increased rapidly beyond the 600-psi pressure range and it was therefore decided to divide the pumping work between high and low boiler feed pumps with intermediate feedwater heating at approximately 600-psi. The power required by the high-pressure pumps was so high that the intermediate heater could not absorb all of the resulting exhaust steam. After a number of attempted solutions, it was decided to resort to a bleeder-type turbine for this drive, allowing the intermediate heater to condense as much steam as it could and exhaust the balance to the 3-psi auxiliary system. In order to maintain 75-psi at full load, the exhaust end of these turbines had to be restricted and this presented difficult problems in the event of a heater failure. A solution was arrived at by installing heaters in duplicate as well as three half-size high and low-pressure boiler feed pumps. Each turbine was so designed that it could carry $^1/_2$ full plant load when operating non-bleeding thus protecting against the loss of both line heaters. This installation has been operating successfully since 1937.

Case B

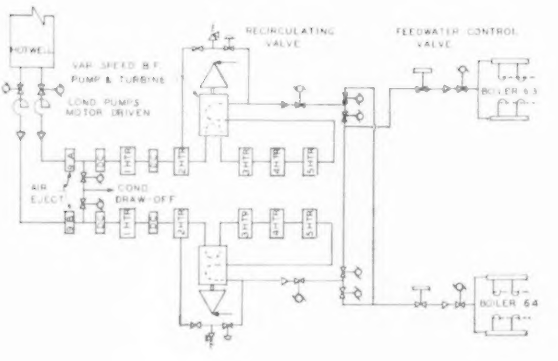
This installation was also a superposed job. The turbines were a 1924 installation with provision for limited bleeding at about 11-psia. The exhaust system was maintained at 1-psi by steam from turbine-driven auxiliaries. There were, however, existing deaerators of the flash type which reduced feedwater temperature to 177 F corresponding to about 7-psia. The feedwater could be heated to about 198 F by utilizing the 11-psi bleed as compared to the 220 F water available in Case A. This meant that the intermediate heater had a greater heat-absorbing capacity utilizing the total turbine exhaust making bleeding unnecessary thereby allowing the heater to find its pressure level at a balance between the available and the usable steam. This worked out fairly well at all loads except very low and very high due to excessive exhaust-steam quantities. For these conditions the exhaust from the boiler feed pump turbine was inducted to a suitable bleed point of a new condensing turbine installed later.

An interesting item in this cycle is the installation of a 250-psi direct-contact heater. This was necessitated by the fact that the eight existing low-pressure boiler feed pumps did not have enough pressure margin for a closed pumping system and the interposition of the "open" heater provided water storage to take care of load fluctuation. While at first somewhat temperamental in operation, stability was soon achieved and it has been operating in excellent fashion. To its credit let it be said that it never failed because of faulty tubes, providing proof that the ancient precept still holds, "what you don't have in a power plant, never causes trouble."

Case C

Shown diagrammatically in Fig. 4 is the pumping system and cycle arrangement associated with duplicate 165,000-kw units installed at Southwark Station. The notable item in this installation is not the three half-size multiple pumping units but rather the utilization of a "house turbine" to provide the energy source for all electrically driven auxiliary equipment such as fans, pumps, and so on. Design for this block of power generation was begun prior to World War II and the paramount consideration was the restarting capability should isolation occur because of possible bomb damage. By employing a turbine-driven generator, exhausting to its auxiliary condenser or atmosphere during a start-up period, permitted isolation of this station from the Philadelphia Electric Company electrical loop system or dependence for electrical energy from an outside source was not required. A self-starting internal-combustion engine and generator were provided to supply the bare minimum requirements for lighting-off the boiler. A fire having been lighted in the boiler, pressure could be increased gradually, the "house-unit" started and auxiliary equipment energized in their order of importance to reach a substantial and dependable condition on the plant.

Normally this generator supplies energy to the auxiliary buses and the turbine exhausts to the 35-psi heater and inducts all excess steam back to the main turbogenerator unit.



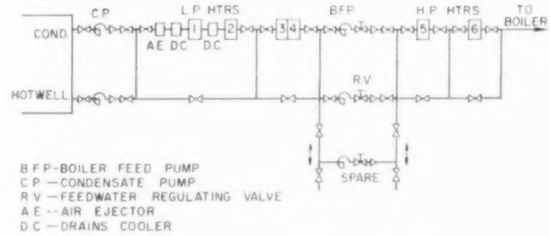


Fig. 6—Delaware and Cromby stations—heat cycle

Fig. 5—Richmond station unit 9—heat cycle

This arrangement proves economical because the house-unit has a sufficient load so as to make it comparable in efficiency to the main unit and actually operates in parallel with it. The full-load pressure ratio of 850- to 35-psi and mass flow when carrying 12,500-kw under steady-state conditions, are sufficient to permit the design of a close-clearance and low-leakage turbine.

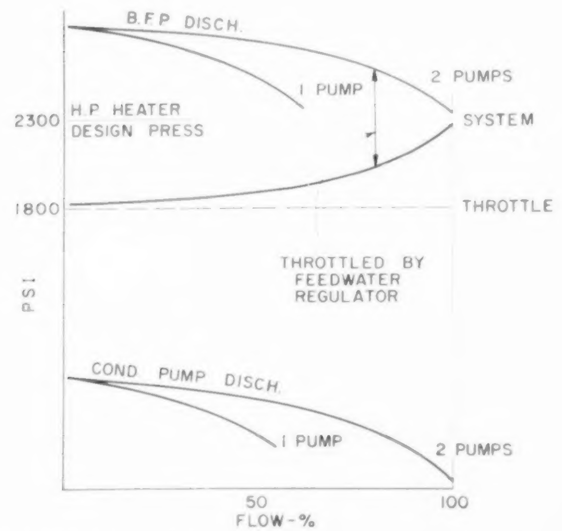
This arrangement, however, is not achieved without hazard and complexity. Extreme caution must be exercised and a maximum reliability in the design of component must be realized in order to prevent overspeed (to destruction) of the main unit in the event of a throttle trip. To this end, an automatic trip valve is installed (as close as possible to the induction point on the main unit) on the house-unit exhaust so that exhaust steam or stored energy cannot produce a hazardous condition of the main machine. This valve must possess the same degree of reliability as the throttle stop valve and is required to act just as quickly. Danger also exists should the extraction valve of a feedwater heater close in which case all the exhaust steam will be inducted back into the main unit rather than being partially absorbed by the feed-heating cycle. If overspeed should occur, the induction and throttle valves close simultaneously.

As previously mentioned, it will be noted that the high-pressure boiler feed pumps in this installation are located after the last heater, thus utilizing the heat input for pumping to the best advantage while at the same time effecting a considerable saving in capital cost of feedwater heaters because of their lower design pressures.

Case D

Fig. 5 shows schematically the 1949 installation of the feedwater cycle employed on a 165,000-kw tandem-compound turbogenerator unit installed at Richmond Station, unit No. 9. A study of this diagram presents two items for consideration. First compare this figure with Fig. 4. Immediately evident is the lack of header valves on the feedwater cycle at the boiler feed and condensate pump suctions and discharges, also a lack of cross-connecting piping. This configuration of equipment minimized the capital investment required for rather costly feedwater valving.

Also it should be noted that only two half-size pumps are installed. These pumps are of the re-entry type. Being turbine driven they may be speeded up thus permitting three quarter load to be carried by a single pumping cycle. As previously mentioned advantage can be made of the two other desirable characteristics of this cycle; namely, relatively low-pressure heater could be purchased and maximum benefit could be



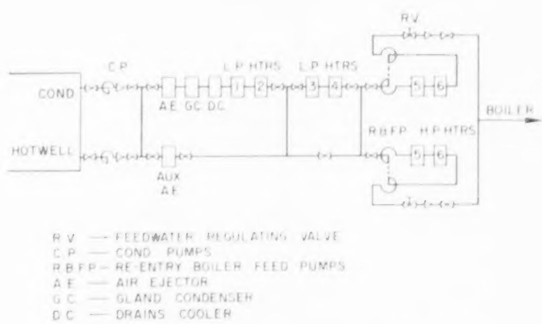


Fig. 8—Schuylkill station unit 1—heat cycle

Fig. 7—Delaware and Cromby stations—system characteristics

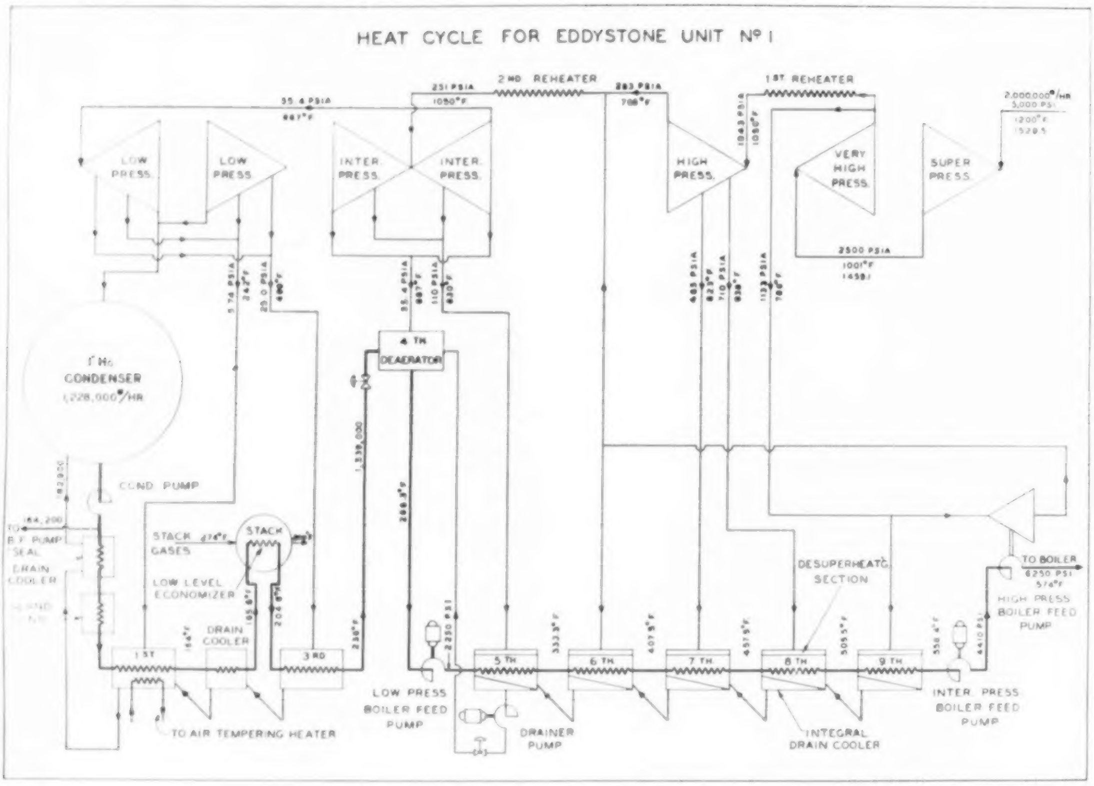


Fig. 9—Heat-cycle layout of the Eddystone No. 1 Unit

derived from the boiler feed pump power input. This was accomplished by locating the final stage of pumping after the highest pressure feedwater heater. A more economical discharge piping and valve design was arrived at by using the variable-speed turbine drive. The necessity was eliminated for subjecting this equipment to the shut-off head of the pump as would have been the case for a constant-speed pump drive.

Case E

Fig. 6 shows the arrangement of the cycle on units Nos. 7 and 8, Delaware Station and units Nos. 1 and 2, Cromby Station, dated 1952 and 1954, respectively. Of concern is the fact that the high-pressure boiler feed pumps are located ahead of the last two stages of feedwater heating. This position in the cycle was brought about by the suction requirements of the pumps (temperature wise) and the cost of feedwater heaters.

It should be noted that the high-pressure feedwater heaters so located with the feedwater regulating valves located ahead of them permitted the use of considerably lower pressure feed heaters than would have been required if the regulating valve had been positioned at the cycle outlet. This latter arrangement would have required a heater design capable of withstanding the "shut-off" head of the high-pressure boiler feed pumps. Fig. 7 clearly shows this difference in design requirement.

The advantages thus realized by this configuration are decreased capital investment in feedwater heaters, piping, valving, plus decreased pumping costs.

Case F

Somewhat paralleling and analogous to the installations at Delaware and Cromby is the unit No. 1 installed at Schuylkill Station operated in June 1958. As indicated in Fig. 8, a re-entry type of pump was again utilized. Advantage could be obtained from this type of cycle because the pumping pressure rise is so divided between the high and low-pressure elements of the boiler feed pump that it was possible to design the high-pressure feedwater heaters for 2000-psi although the throttle pressure is 2400-psi. This pressure of course is sufficient to sustain "shut-off" head of the low-pressure element of the feed pump, normal operating pressure being approximately 1600-psi, a safe margin over 340 F suction temperature even in the most conservatively designed plant. The high energy input to the high-pressure pump element is balanced by the return in heat to the feedwater while advantage is taken of the power saving in pumping colder water in the low-pressure element of the same machine.

Of significance in this re-entry pump is the fact that by careful arrangement of stages it was possible to obtain first-stage suction pressure on both ends of the shaft thus facilitating a rather simple method of sealing and leak off in a packless-gland assembly.

Case G

Fig. 9 shows the heat-cycle layout of the most recent and largest turbo-generator unit on the Philadelphia Electric Company system, Eddystone Station unit No. 1. Designated as a supercritical-pressure

unit, it has throttle conditions of 5000-psi and 1200 F with two stages of reheat. The term critical pressure may be defined as the saturation pressure corresponding to a temperature above which a gas cannot be liquefied by compression alone. At this critical state point the specific volume of the gas and the liquid are identical and, at a temperature in excess of the critical, it is impossible to have a mixture of vapor and liquid. The values identifying this state for water are 705 F and 3206-psia. Fig. 10 graphically shows the specific volume-temperature relationship for steam and water.

It will be noted on this curve that as temperature increases the saturated water volume increases and the specific volume of steam at saturation pressure decreases as the temperature increases until at the critical point the specific volumes of both are identical.

A companion curve, Fig. 11 has been included to show the temperature-enthalpy relationships for the media.

It will be recalled that on a T-E chart, pressure is illustrated as a horizontal line under the saturation line. This line also may be thought of as a graphical representation of the latent-heat value. Therefore, examination of the saturation line will make it evident that as pressure increases the latent heat decreases until a critical state point (3206-psi and 705 F) exists, at which point the latent heat is zero. Both of these conditions existing at the critical point are subjects for serious consideration in the design of the boiler, turbine, and auxiliary equipment for the feedwater cycle. The steam-generator and turbine-design considerations have been described previously in papers presented by the respective manufacturers, Combustion Engineering and Westinghouse Electric Corporation. However, an enumeration of the requirements and qualification imposed by operation at or above the critical state point, is in order to lay the ground rules, so to speak, in the selection of equipment to satisfy the pumping requirements.

One obvious requirement for operation in the critical range is that the fluid pressure never be allowed to fall below 3206-psi. If such a condition did exist localized boiling would occur. Because of the change in specific volume with changes in pressure, serious unbalanced flow might be experienced in the steam-generator tubing; i.e., in the waterwalls or superheater circuits. As a result, flow could not be regulated sufficiently well in the individual paths; thus overheating and/or burn out could occur in this once-through type of steam generator.

Another basic requirement of this steam-generator is that under no circumstance should the flow through the unit be allowed to drop below approximately 30 per cent of 2,000,000 lb per hr, the full-load flow on this machine.

Although the rated turbine-throttle conditions are 5000-psi, 1200 F, provision also must be made to operate the boiler-turbine unit as a 3500-psi machine up to approximately three quarter load then as a variable pressure unit up to 5000-psi throttle pressure level.

Arrangement of Heaters and Pumps

A suitable pumping system to meet the above specification, together with supplying the large feedwater flow at extremely high pressure, was devised to produce the maximum flexibility and dependability. Fig. 12 shows the arrangement of feedwater heaters and pumps in the Eddystone unit no. 1 cycle.

In order to handle the very large quantity of feedwater, two parallel circuits were employed. Each circuit will handle one-half capacity at any load up to the 5000-psi throttle pressure when operating in parallel. If a fault should occur in one line of heaters when at full load, the other line will have a rating of approximately 60 per cent at a throttle condition on the main unit somewhat less than the 5000-psi rating.

To meet the very high pressure at the economizer inlet, the use of two or more stages of pumping was required. Three stages were finally settled on with the high-pressure feedwater heaters being placed between the low and intermediate-pressure boiler feed pumps. By positioning the heaters thusly, and dividing the pump pressure rise into about three equal steps, the feedwater heaters could be purchased for 3000-psi, a reasonable low pressure considering the 6300-psi existing at the high-pressure pump discharge. Several other installations on the Philadelphia Electric Company system at present are operating successfully with 2300-psi designed heaters and it is contemplated that satisfactory results will be produced here since the full-load

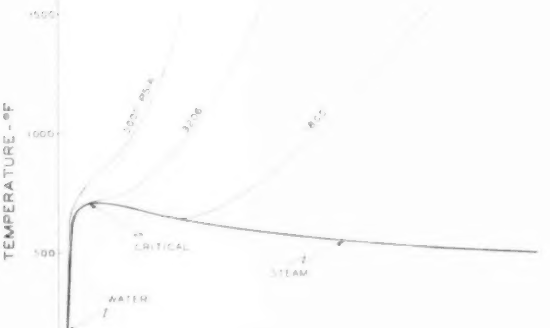


Fig. 10—Specific volume-temperature relationship

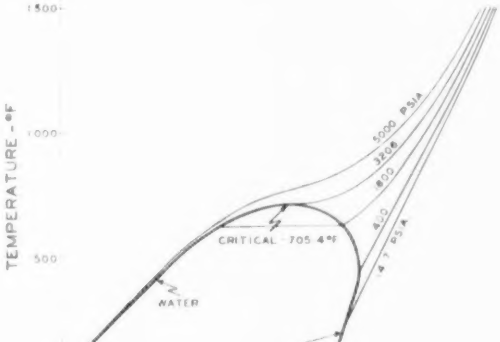


Fig. 11—Temperature-enthalpy relationship

channel pressure will not exceed approximately 2400-psi.

With the intermediate and high-pressure boiler feed pumps located after the highest pressure heaters, they will be handling water at an elevated temperature. While this increases the total pumping power substantially, most of this energy is returned to the feedwater in the form of heat. At full load, the increase in enthalpy of 18 Btu results in a substantial gain of cycle economy, more than overcoming the higher power requirement coincident with pumping hot water.

Boiler Feed Pump Drive

The next decision to be made was the manner in which the boiler feed pumps were to be driven. Several systems or combinations suggested themselves. The system that finally proved most efficient and economical was to utilize motor drives for the low and intermediate boiler feed pumps and a turbine drive on the high-pressure units.

Recalling the characteristics of a constant-speed motor-driven pump, it will be recognized that two conditions occur which had paramount bearing on the consideration of turbine drives for the high-pressure boiler feed pumps. A constant-speed pumping unit will have a continuously rising head characteristic toward shut-off. This fact together with the change in specific gravity of the water due to temperature change over the turbine load range will produce a very high shut-off pressure. It is evident therefore, that the boiler feedwater piping between the high-pressure pump discharge and the feedwater regulating valves would have to be designed not only for the accumulative friction drops throughout the system but also for the shut-off conditions produced by the pumping units. While these characteristics are undesirable at the highest pumping level, they were used to advantage in the design criteria of the low and intermediate units as will be described later. Fig. 13 shows the overall pumping and system characteristics. If a constant-speed unit had been employed the shut-off design pressure on the discharge piping would have been approximately 9500-psi rather than 6300-psi which was effected by the utilization of a variable-speed mechanism. Since this feed piping is of the forged and bored variety a considerable monetary saving was realized.

Bearing in mind the fact that this turbo-generator unit will have the best heat rate of any on the Philadelphia Electric Company system it is expected to operate at full load as much as possible. The choice of a variable-speed drive therefore should be one which would not effect the overall efficiency of the cycle. Several types of variable-speed producers were considered for use with constant-speed motor drives. All such machines produce inherently a parasite loss which has to be evaluated not only throughout the entire operating range but what is more important, at peak loading a constant loss would have to be tolerated with a deleterious effect on the plant efficiency. A variable-speed coupling, magnetic, fluid or mechanical, falls into this category. Investigations relative to boiler feed pump types, indicated that a machine, small in physical size, would be very desirable for many reasons,

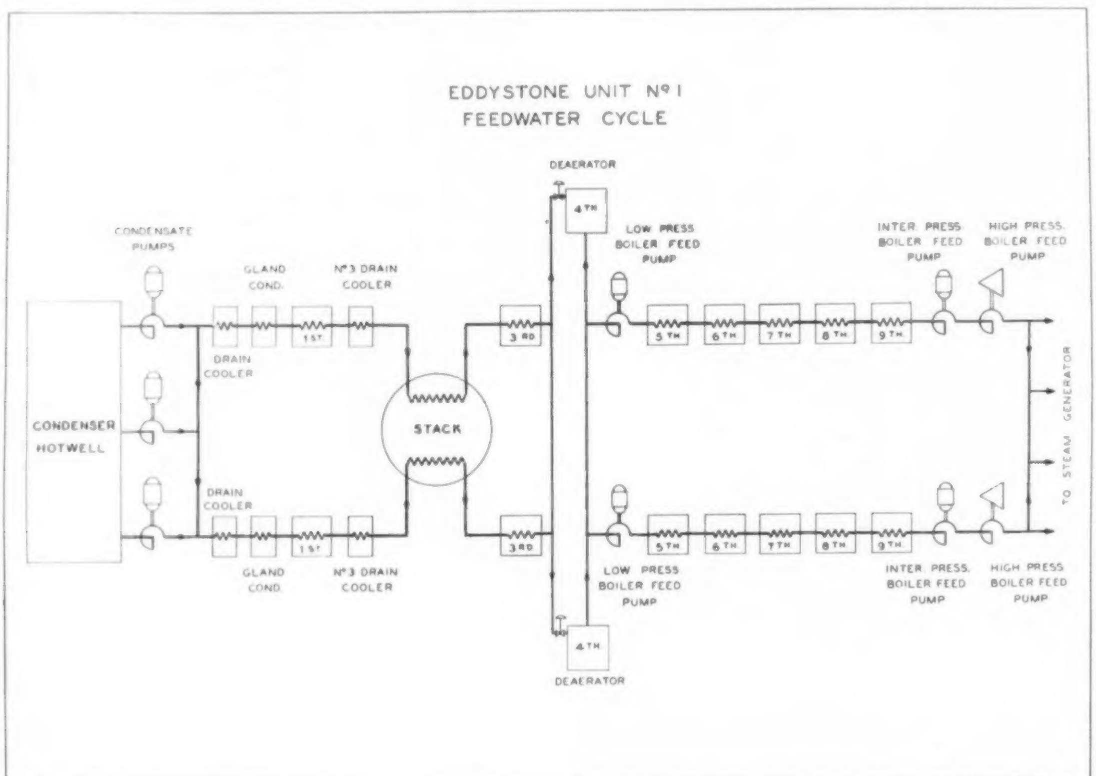


Fig. 12—Feedwater heaters and pumps in Eddystone No. 1 Unit

among which are shorter bearing span, less staging, thinner shaft and less metal thickness throughout which is inherently required by the extremely high pressures involved. Coincident with small pump size is higher rotating speed which in turn necessitates the use of a step-up gear with its complexity, losses and questionable reliability for a continuous availability such as is mandatory for this system.

Turbine Drive Selected

The requirements as outlined were a "natural" for a turbine drive on the high-pressure boiler feed pump. Many installations of this type have proven dependability records both on our own and other utility systems.

Investigations were made to determine which locations would be the most economical for obtaining source steam and disposing of the exhaust and which type of turbine would be most flexible and reliable for the service. Recalling that the main turbine is a double reheat machine, studies indicated that the proper location for the boiler feed pump turbine drive was operating between the No. 1 and No. 2 cold reheat Lines. A satisfactory turbine could be designed for this location because the full-load inlet and exhaust pressures would be approximately 1100-psi and 325-psi, respectively. This location also exhibited another very desirable characteristic; namely, of holding a fairly constant pressure ratio over the entire range during which it is contemplated the machine would be used. Reference to Fig. 13 shows that expected operation range for this turbine drive will be from 30 per cent to full-load boiler flow. As explained later the 3600-rpm constant-speed motor-driven low and intermediate boiler feed pumps will satisfy the system requirements below the 30 per cent load (or flow) condition. It also should be noted that more than adequate energy is available between the No. 1 and No. 2 cold reheater points at all loads above the 30 per cent minimum pressure level to satisfy the horsepower demands. An efficient machine was made possible also by reason of the fact that large machines, developing approximately 4000 hp each, at 5000-rpm were required to meet the system demands.

Several types of mechanical drive turbines were studied: Bleeding with one or more extraction points for feedwater heating; condensing; emergency condensing and/or atmospheric exhaust, and so on. The most economical proved to be the nonbleeding back-pressure type.

Crowding in the staging because of the high volumes of steam required to develop the required power over the full range of operation plus the volume of steam required for extraction to one or more heaters made the bleeding type of machine unattractive. When an emergency condition of high water in feedwater heaters which caused a trip of the extraction valves was considered this type of unit appeared impractical.

As the system evolved, a feed heater is located at the exhaust of this machine and will preferentially use the exhaust steam at a low enthalpy for feedwater heating rather than the rich cold reheat steam from the main turbine. In this fashion, the excess exhaust only is injected into the No. 2 cold reheat and a trip out of the heater because of high water will have very little effect on the pressure ratio across the boiler feed pump turbine

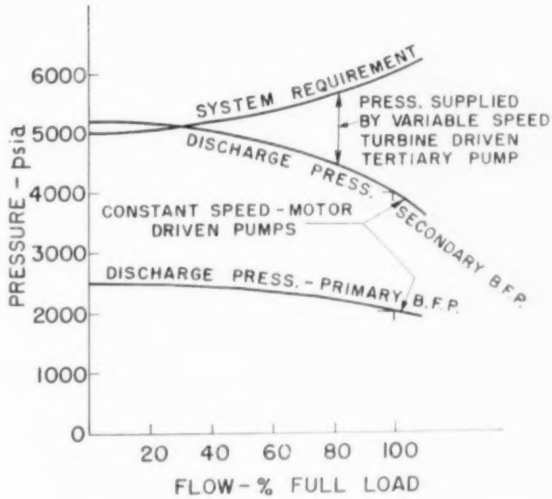


Fig. 13—Eddystone station unit 1—system pumping characteristics

drive. An atmospheric safety-valve exhaust for limiting pressure at this location and a manual bypass are also provided for use when overspeed testing with pump either connected or disconnected and for turbine washing operations.

Special Requirements

As mentioned before the inherent characteristics of a pump curve to rise from full load to shut off and the change in specific gravity of the water due to temperature change were used to fulfill two other requirements associated with this supercritical steam-generator unit. These are: At no time should the flow be allowed to go below 30 per cent through the waterwalls and at all times the pressure must remain above critical.

Again referring to Fig. 13, it will be noted that the motor-driven low and intermediate boiler feed pumps will develop sufficient pressure to keep the water above critical pressure and satisfy the 30 per cent flow figure even while a throttle pressure of 5000 psi is being maintained. At any intermediate throttle pressure down to and including 3500 psi these two pumps (low and intermediate) working either in a single or parallel line will assume a greater proportion of the pumping load depending on the system curve characteristics for the throttle condition selected. Therefore the foregoing requirements are satisfied by proper design selection of pump capacity and the characteristic head rise of the constant speed units.

Thirty Per Cent Flow

Perhaps an explanation of the 30 per cent flow requirement is in order at this time. With the supercritical steam generator during start-up a system of bypassing is used in order to increase the firing rate of the steam generator which in turn produces the desired throttle temperature of 1000 to 1200 F as considered most economical and suitable for the pressure and rate of heating of the many thick-wall parts involved (piping and turbine, etc.) Interlocks of course are provided to permit starting when flow and pressure are satisfied together with requirements for trip out or run back on

partial or complete loss of electrical generator load. A system of interlocks also has been developed for both the water piping on the suction of the pumps and on the oiling systems to prevent starting the pumps or to trip them out should a malfunction occur in either the water or oiling system. This requirement is dictated by the fact that either the intermediate or high-pressure pump may act as a hydraulic turbine during start up, shut down or when load is being carried by the low and/or intermediate boiler feed pumps.

Perhaps extensive discussion was devoted to the system employed on unit No. 1 Eddystone Station but it is felt that this design embodies a great deal of thinking and represents possibly the ultimate in simplicity for the very complex system requirements. Certainly the combined efforts and resource of all parties involved, both the manufacturers and the utilities alike, were required and generously contributed in order to solve the many new and different problems involved with such an undertaking.

Future Considerations

To predict the location of the boiler feed pumps in the heat cycle in a few years hence would be an impossible task. However such factors as the increased reliability of the pumping machine brought about by development of better alloys to withstand the corrosion, erosion action of water; higher rotating speeds to permit the design of physically smaller units for comparable flows; the increased faith designers are placing in single rather than multiple or spare pumping units; and above all the constant demand for more economical systems has constantly brought pressure upon designers to resort to ingenious schemes to achieve their goals.

The constant increase in main unit temperature, pressure and capacity has been a challenge to the entire industry. Pump manufacturers and designers have kept pace and thus far have been capable of meeting this challenge. The boiler feed pump continues to remain "the heart of the cycle."

23rd National Exposition of Power and Mechanical Engineering

A BRILLIANT industrial panorama has been created for the 23rd National Exposition of Power and Mechanical Engineering, at the New York Coliseum, December 1 to 5. Forecasting the business expansion predicted for 1959, it will portray the methods and mechanisms whereby the manufacturing industries will be able to increase productivity with economic advantage.

The Exposition will be held under the auspices of The American Society of Mechanical Engineers in conjunction with its 1958 Annual Meeting. It will be open to members of the Society and guests associated with the power and related mechanical fields.

Continued progress in the engineering of power generating plants has produced a number of technical improvements since the last Power Show in the Coliseum, two years ago. Applications of power in manufacturing and in service facilities of every description have multiplied and improved to a noteworthy extent. Displays of some 300 exhibitors in this nation-wide review will demonstrate the high potential in new and improved equipment that is ready and able to serve industry.

Basic advances in structural materials are responsible for many profitable changes in design that will be disclosed at the Show. A number of innovations in the metals have recently come into the picture, while spectacular advances in non-metallics have made important contributions to the improvement of machinery and processing equipment. These influences are revealed in newly released designs that are characterized by more effective stress distribution, improved resistance to contamination and wear and, in many instances, weight reduction as well. More discriminating selection of materials, more dynamic draftsmanship and modernized styling and finish will be evident in many categories.

Moreover, research and development are continuous processes. Producers whose recently new materials are featured in equipment of current design are expected to announce a number of still more recent advances.

For instance, a new variety of wrought iron will be offered as affording 25 per cent greater resistance to corrosion than any other wrought iron specification. It is termed the first change in this material in over 20,000 years. The same producer will also introduce a new line of polyvinyl chloride pipe and sheet.

Another challenge is offered by a producer of stainless and high alloy steel tubing in advocating the use of stainless to overcome corrosion and erosion problems in main surface condensers "in both fossil fuel and nuclear type generating stations."

At the same time, producers of the more familiar standard materials are offering increased assortments of specifications and shapes. The variety is suggested by the partial array of a leading line of copper and brass which includes long length, dual gage, U-bend condenser tubing, condenser tube bundles, drilled tube sheets, bus bar and tube, conduit, commutator bars and segments, instrumentation tubing, capillary tubes, and flexible metal hose, both seamless and strip wound.

One of the headline displays of material will be the first show-time presentation of a new acidproof insulating refractory consisting of a lightweight foam of 99 per cent pure silica, which has non-interconnecting bubbles, and therefore applicable to many uses where sealing, as well as thermal insulation, is required.

This newest member of the glass family consists of silica boiled at over 3100 F with a carbon foaming agent. It compares in density, at 10 to 15 lbs per cu ft, with carbon black (97 lb/cu ft), and acid brick (140 lb/cu ft). It is reported to have good dimensional stability, not to warp, shrink, slump, crack or spall under thermal shock; and with 130–210 lb/sq in. compressive and 120–150 lb/sq in. flexural strengths, to have adequate load-bearing possibilities.

A familiar glass insulation is now being prefabricated for valves, fittings and flanges by new methods. Its applications also include pipe coverings, electrical products, filters and reinforced plastic applications.

By IGOR J. KARASSIK*

Worthington Corp.

The boiler feed pump and its associated equipment represent a major operating and maintenance consideration in today's power plant. Here we run in question and answer form a series of clinic sessions on various boiler feed pump problems. The replies are the work of one of the topmost pump authorities and give specific information which we hope will prove valuable to our readers.

Steam Power Plant Clinic—Part IV

QUESTION:

Our new 25,000 kw unit is served by two 100 per cent capacity boiler feed pumps which are duplicates of pumps installed in an older station. Recently we had noted a very rapid increase in power input to these feed pumps and, in our concern over the possibility of premature wear, we had them opened up. No internal leakage was reported, nor were the clearances markedly increased. The rotor was cleaned up and the pumps reassembled. After reassembly, the pump power consumption returned to normal. What could have been the cause of the increase in power we had noted and why did this disappear after the pumps were opened up? We did not replace any of the wearing parts.

ANSWER:

While you do not mention the appearance of the pump parts when the pump was opened up, you do state that "the rotor was cleaned up." I imagine that your maintenance mechanics found the impellers heavily coated with oxides and brushed them or sandblasted them, depending on the degree of adherence of the oxides to the impellers.

Caution should be exercised in the interpretation of field tests results which show evidence of premature reduction in the pump net capacity or of a rapid increase in power consumption. The first conclusions reached generally are that internal clearances have increased to the point that a complete overhaul is required. Another possibility contemplated is that serious internal leakage is taking place between the stages. However, in a number of cases, these conclusions prove to be erroneous.

While the use of chromium steels for pumps parts and for pump casings of boiler feed pumps is quite effective in resisting corrosion-erosion attack, there still remains the problem of oxide formation in equipment located upstream of the feed pumps. Various iron compounds are formed in deaerating heaters, feedwater lines, heater drip systems. These compounds are quite readily deposited on the internal walls of the impeller waterways as well as on the casing walls and external impeller walls.

Normally, the reduction of the flow areas due to these deposits is negligible. However, a few extreme cases have in the past come to my attention, where up to 0.040-in. or even 0.060-in. build-up of dark brown oxide had formed on each internal impeller wall in impellers of approximately $^3/_4$-in. width. The reduction in area in such cases amounted to between 10- and 16-per cent. Obviously, such reduction in area is accompanied by an equivalent reduction in pump capacity.

The pump efficiency suffers doubly: first from the reduction in capacity and second from the fact that the resulting profile of the impeller becomes distorted from its ideal design shape and the hydraulic efficiencies of the impeller and of the pump are unfavorably affected.

The iron compounds deposited on impellers vary from very soft consistencies which can be removed with a wire brush to extremely hard, brittle compounds which generally adhere quite strongly to the impeller walls and which need to be sandblasted. These latter compounds are dangerous for another reason, that is because of the hardness of any broken-off particles. If these lodge themselves within close internal clearances, they may cause damage to the pump.

Unfortunately, there is no ready means available to predict the possibility of such a build-up, nor is there any assured and safe method developed to remove it without dismantling the pump. On the other hand, there are a number of feedwater treating methods to reduce or eliminate this condition once it has been detected. While the problem of feedwater treatment is extremely complex and each case should be thoroughly analyzed on its own merits, I may list a few of the methods that I have seen employed. In one particular case, a program of dosing the make-up water with a slight amount of ammonia was used. Addition of concentrated boiler water to the deaerator is frequently successful. Since dosing with sulfite for oxygen elimination frequently leads to iron compound build-up, a different method of oxygen elimination will generally reduce fouling of the pump internals.

But once such fouling has been detected, the proper procedure is to institute a thorough study of the problem by a feedwater treatment specialist.

* Assistant to Vice President and Consulting Engineer, Harrison Division.

Question:

Our boiler feed pumps are provided with recirculation by-passes for thermal protection at light loads. The pumps are designed for 400 gpm and 900 ft total head and are driven by 150 hp, 3560 rpm motors. The recommended minimum flow is 20 gpm and it is provided by means of an individual by-pass line from each pump, ahead of the check valve, and returning the deaerating heater at the pump suction. This line has in it a $^7/_{32}$-in. orifice 4 in. long, drilled in a piece of stainless steel. Because of the expense of an automatic control, we have installed a manually controlled valve to open and close the by-pass. The orifice is followed by an elbow in the return line. We find that the high velocity of the water issuing from the orifice erodes the elbow in less than two months. What can be provided to eliminate this difficulty?

Answer:

The description of the installation indicates that great care has been taken to protect the pump against overheating when operating at light loads. By rule-of-thumb methods, the amount of by-pass provided would limit the temperature rise to 15 F which is generally the recommended value. However, the piping of the recirculation line can readily be improved. The orifice should first be followed by a length of straight pipe of at least 12 in. to 18 in. After this, a Tee should be provided, as shown on the attached Fig. 1, to lead back to the heater. The Tee should then be followed by another length of straight line, terminating in another Tee or pipe coupling fitted with a stainless steel plug. It is this stainless steel plug which will have to take the brunt of the high velocity stream coming through the orifice. The plug should have a very long life. At worst, it may require replacement, but it is cheaper and simpler to replace it than an elbow in the recirculation line.

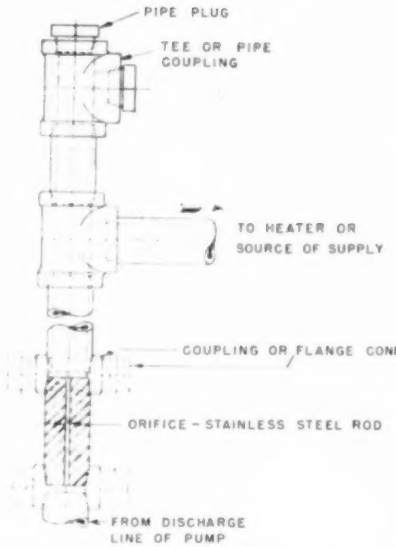
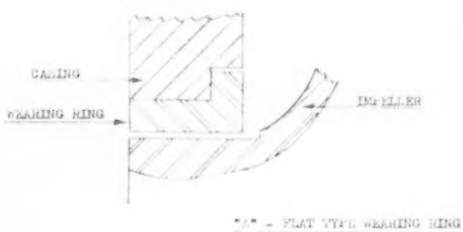
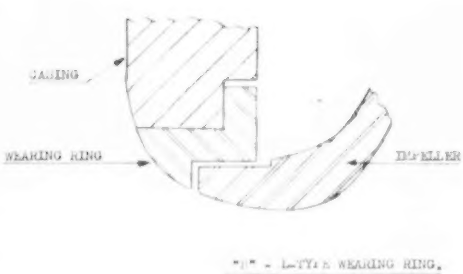


Fig. 1—Suggested proper orifice and piping arrangement calls for at least 12 in. to 18 in. of straight run between orifice and a tee take-off to the heater, plus another run of straight pipe to a second tee fitted with a stainless steel pipe plug.





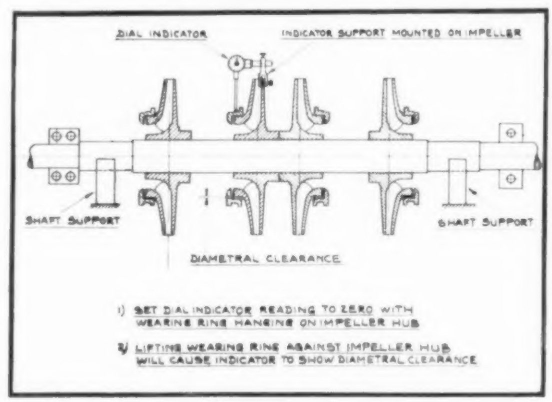


Fig. 2—Two different types of wearing ring construction for a pump show that the flat-type, "A" left, permits a feeler gage to check on wear. The L type, "B" left, requires the aid of a device such as Fig. 3

Fig. 3—A dial indicator mounted on the impeller shroud and giving readings such as described in the illustration yields some information of wearing ring clearances

Question:

Our maintenance crew has found that the inspection of some of our multistage axially-split casing centrifugal pumps can be carried out without stripping the rotor. The clearances at the wearing rings are measured by inserting a feeler gage between the rotating and stationary parts. Some other designs, however, are such that a feeler gage cannot be used because the stationary wearing ring has a lip

which overhangs the front part of the impeller hub. Is there any short cut method which would permit us to check clearances without completely stripping the rotor?

ANSWER:

I presume that in the first case you refer to a construction similar to that shown on "A," Fig. 2 where the leakage joint is flat and therefore permits the insertion of a feeler gage. Wearing rings of the L-type, shown on "B," Fig. 2, of course prevent the insertion of the gage. It is still possible to obtain a reasonably accurate check of the diametral clearances in the following manner:

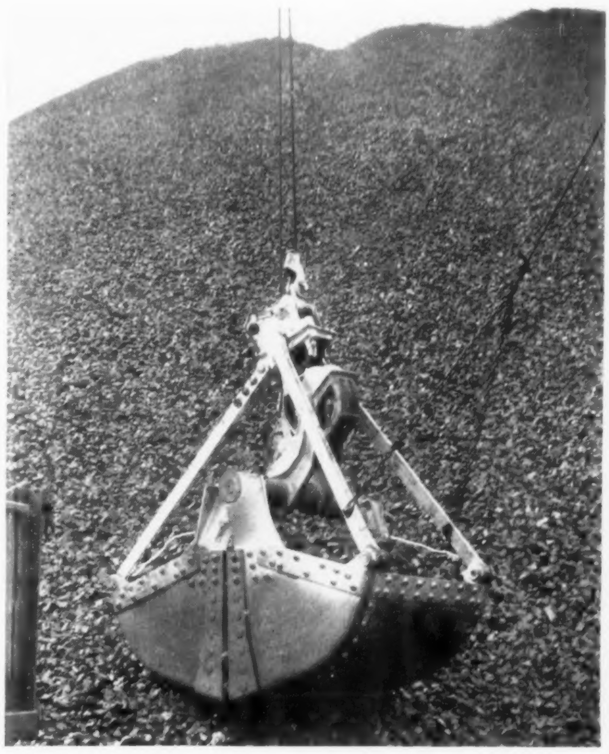
1. Mount a dial indicator on the impeller shroud (as in Fig. 3) and with the stationary ring resting on the impeller wearing ring hub, set the dial reading at zero.
2. Leaving the dial indicator in position, push up on the stationary ring from below and make a record of the maximum dial reading. This corresponds to the diametral clearance.
3. Repeat this operation for every clearance joint and make a record of all readings.

Obviously, if these readings indicate that the clearances have not changed appreciably from the initial dimensions given in the instruction book for the pump in question, no further dismantling of the rotor is necessary.

One note of warning, however: this short-cut method will disclose the extent to which clearances have increased, but will give no indication as to the condition of the adjacent clearance surfaces. In other words, the presence of burrs, grooves or indentations caused by the passage of foreign matter through the clearances and the resulting damage to the surfaces will go undetected.

Utilities Join With Coal People

A program sponsored by the Association of Edison Illuminating Companies and Edison Electric Institute, representing the investor-owned electric utility companies, and Bituminous Coal Research, Inc., representing the coal industry, to be conducted during a two-year period beginning this summer, will be undertaken in the laboratories of Bituminous Coal Research, Inc., the national research association for bituminous coal. The research organization will attack both ends of a problem involving the reduction of sulfur dioxide and other contaminants in flue gas as produced and the reduction of sulfur in steam coals prior to burning. At present, methods for removing contaminants from flue gases are economically unattractive. The research program will be supervised by a Joint Research Advisory Committee, yet to be formed.



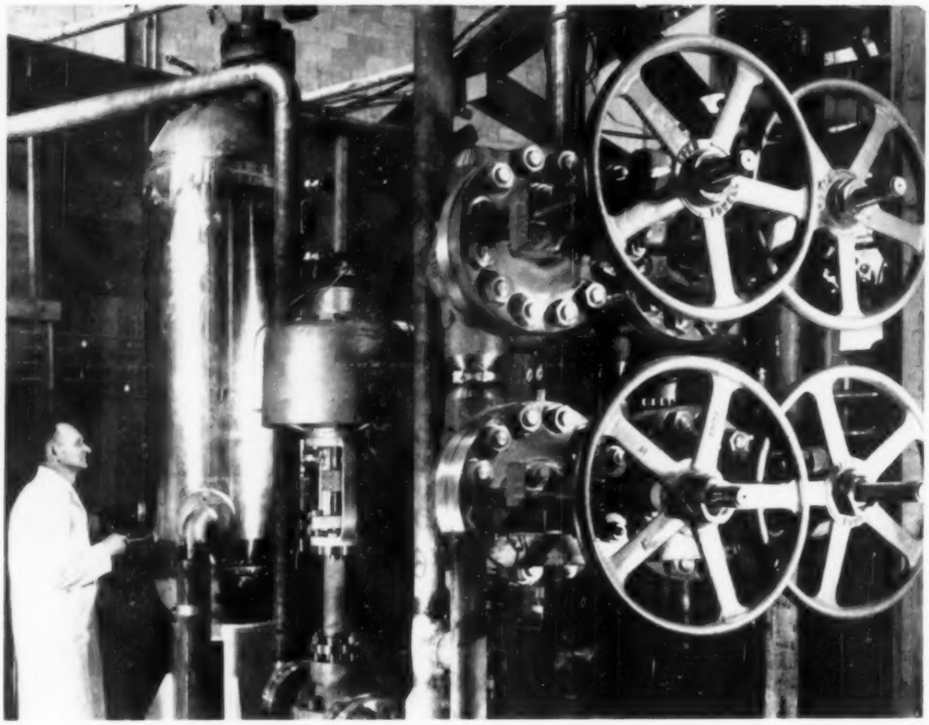


Fig. 1—Off-site or out-of-pile test facilities to enable checking all the process control and instrumentation of a water loop for the Idaho Falls, Idaho, reactor had to be fitted into the cubicle area, above

Test Loop Probes Inner of New Reactors

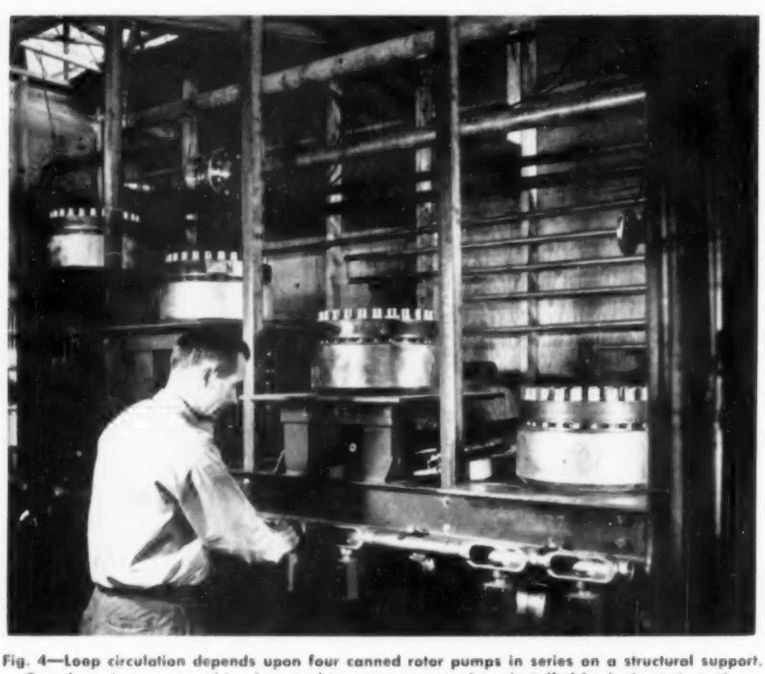


Fig. 4—Loop circulation depends upon four canned rotor pumps in series on a structural support. Cans have been removed in photo and temporary cover plates installed for hydrostatic testing

THE new Engineering Test Reactor at Idaho Falls, Idaho, is expected to supply a range of answers on reactor fuel samples placed in an in-pile test section for study under controlled temperature, pressure and radiation flux. While the Test Reactor was still in the erection stage it was decided to develop and set up a test loop, off-site, upon which the functions of all process and control equipment, the instrumentation systems and the safety interlocks and alarms planned for the Test Reactor could be checked out. The test facility consists essentially of a pressurized water heat exchange system and demineralizing equipment. This off-site test facility, further, was to be run through its paces at the site of its initial assembly and then dismantled and shipped by rail to Idaho Falls in a manner that permits ready re-assembly at the ETR site.

The Lummus Company Engineering Development Center, Newark, N. J., was awarded the sub-contract by Knolls Atomic Laboratory for this off-site test loop and delivery was made at the Idaho Falls site during the summer months. Lummus in its test loop simulated the confinements of the area assigned to the facility. As a direct result the major loop components, Fig. 1, were crowded into the confines of a primary cubicle which extends as a wedge from the reactor base. Because

Fig. 2—Electrically heated pressurizer maintains loop pressure above the vapor pressure of the loop water and hence controls heater operation through a saturable reactor

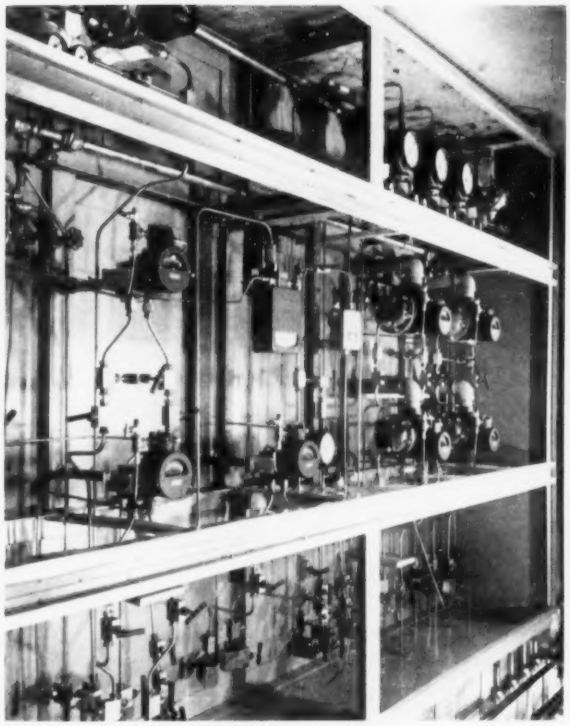


Fig. 3—Sensing lines from pressure tops and flowmeters meet at an instrument cabinet in the secondary cubicle with sliding gear doors to protect personnel

it will be necessary to get at the reactor at almost any time the designers laid out the major loop components within the relatively small are forming this wedge. In Fig. 1 you will note the emergency cooling block valves are built for extension operators to protrude through a 24-in. magnetic block shielding wall and the clearances are such that the valve handwheels will overlap. The test loop flow, incidentally, is controlled by the 3-in. pneumatic valve in the center of the photo.

Four canned rotor pumps, Fig. 4, in series maintain the test loop circulation. The stepwise arrangement permits aligning the discharge of one pump with the suction of the next. The main loop piping to and from the reactor runs above the four pumps. The inlet piping can be distinguished in Fig. 4 by the stainless steel ring joint flanges. These flanges carry a Dall tube flowmeter between them. Plug-in receptacles for thermocouples and conduit for lead wires are mounted directly on the structural steel.

The pressure within the loop must at all times be above that of the vapor pressure for the water leaving the canned rotor pumps. This requirement is met by the electrically heated pressurizer, as it is called, showing in the left background of Fig. 1 and featured in Fig. 2. Four external heaters maintain temperature within the pressurizer and water level is held by pumping demineralized water into the main reactor loop. There are low and high level alarms and automatic cutoffs built into the system. The pressurizer vessel is centrifugally cast of $2^{1}/_{2}$-in. thick stainless steel with spherical heads.

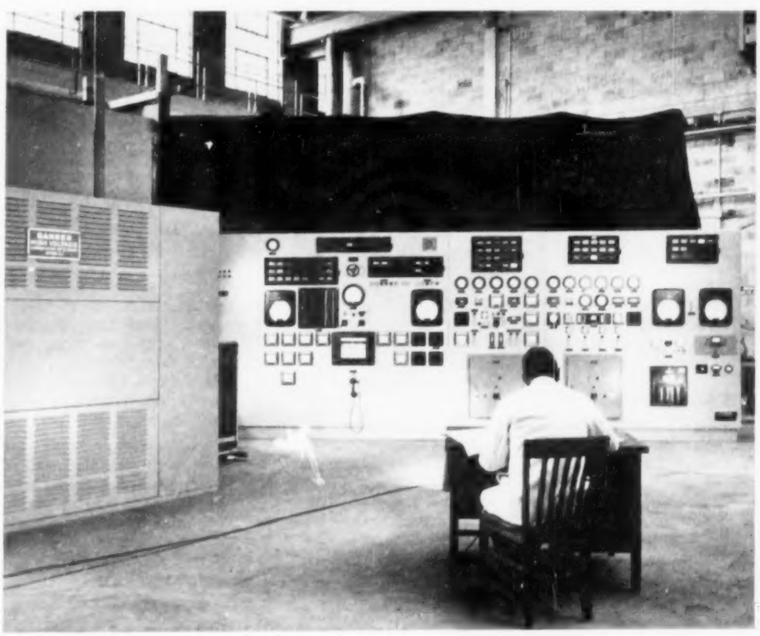


Fig. 5—Signals from primary transmitters near the loop proper send signals to the 22-ft prefabricated instrument panel, above. Serious alarms "scram" the reactor

Fundamental Concepts of Incremental Maintenance Costs*

By D. B. Zelenka† and R. H. Travers‡

Ohio Edison Company

The Ohio Edison System, which serves an area of approximately 8900 sq miles in sections of central and northeastern Ohio and western Pennsylvania, has eleven steam electric plants containing generating facilities of various ratings. This variety of equipment and a two-to-one relationship in coal costs along with a widespread transmission system, the authors believe, dictates a careful study of costs of delivered power.

THE importance of the evaluation of incremental maintenance is increasing with larger generating stations. It is possible to make improvements in the heat rate of units as size, pressures and temperatures increase. Likewise, it is possible to reduce operating costs through centralized control and automation. However, it is not possible to effect the same kind of reductions in maintenance costs. This is due to the fact that a power plant is essentially a heat transfer surface, or wearing surface, and that surface increases directly with the size of the unit. Therefore, presuming no improvements in methods of cleaning and repairing these surfaces, the man-hours required would increase almost directly in proportion to the size of the units. Maintenance engineers have done an excellent job of holding these costs to a minimum by improving methods of maintaining equipment. There are, however, limits to the broad application of these improved methods due to the difference in equipment brought about by technological advances. Furthermore, automation and more complex steam cycles add to the maintenance expense due to additional instrumentation and controls, along with more heaters, fans, and auxiliary equipment.

Maintenance Expense

To illustrate this point, we will investigate one particular plant on our system. (See Table 1.) As a three-unit plant, the relative distribution of costs is shown in Column A; with the addition of two large units, the relative distribution is shown in Column B.

Another study of production costs of the Ohio Edison Company in 1957 compared to 1951 reveals that operating cost (excluding fuel) increased 20.5 per cent while maintenance cost increased 23.3 per cent. This also demonstrates that maintenance expenses are increasing more rapidly than operating costs.

Incremental Maintenance Cost

The increase in maintenance expense is inevitable and with the emphasis now being given to economic dispatch of power systems, the development of methods for determining the incremental maintenance cost is of increasing importance.

The evaluation of incremental maintenance has long been the enigma of any carefully developed economic dispatch program. There is general agreement that maintenance costs are not totally incremental, nor are they entirely fixed. The curves in Fig. 1 illustrate typical cost relationships and their effect on the increment.

It is impractical to determine maintenance increment from a curve showing total or average maintenance cost versus output for any specific period of time, since any

TABLE 1 MAJOR ELEMENTS IN POWER COSTS

	A Three Units	B Five Units
Per Cent	1953	1957
Fuel Cost	79.3	79.0
Operating Cost	10.7	9.9
Maintenance Cost	10.0	11.1
Total	100	100
Operating Cost	51.8	46.9
Maintenance Cost	48.2	53.1
Operating, Maintenance Excluding Fuel	100	100

* Originally scheduled for presentation at the AIEE Fall General Meeting, Pittsburgh, Pa., October 26–31, 1958, as Paper No. CP 58-1308 but cancelled because of the hotel strike. Original title was "Fundamental Concepts of Incremental Maintenance Costs as Used by Ohio Edison Co."

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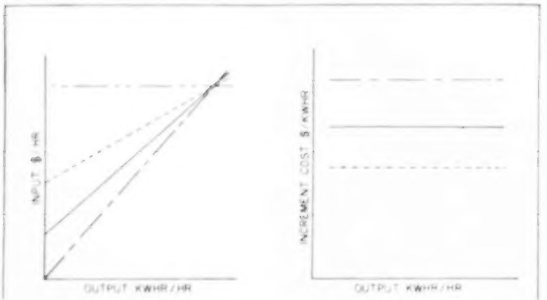

Fig. 1—Typical input-output cost curves and their effect on decrement cost curves

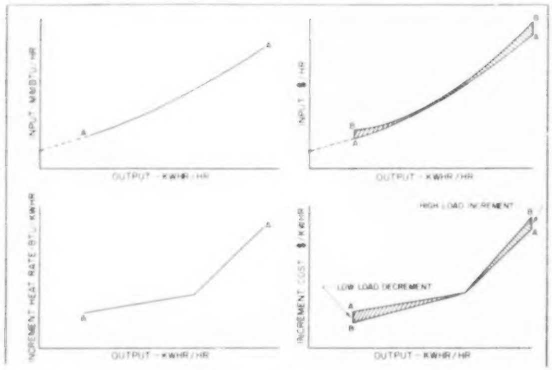

Fig. 2—High and low load effects on input-output data and incremental data

data of this nature gives a sporadic pattern. Maintenance costs are deferred expenditures which do not occur simultaneously with the conditions that produce them, thus accounting for the sporadic pattern.

The actual value of incremental maintenance will also depend on the policies and practices followed by a particular company. Among those factors affecting this cost are: (1) Unit size relative to system load. (2) Overtime practices. (3) Scheduled maintenance program. (4) Manning policy. (5) Contract work.

All these factors, and more, have a significant effect on that portion of maintenance expense which can be considered fixed and that which is incremental. Furthermore, the practices in a particular plant may (and should) change with time, as the demands and load factor on that plant change. For example, as the load factor on a particular plant decreases, one can no longer justify as many full time maintenance men. This reduction in force reduces the fixed charges and may, through overtime, produce labor charges which can be considered as incremental.

From the foregoing, it would appear that it is impossible to recommend one solution of maintenance increment which would be applicable to all situations. We can, however, outline some of the factors which affect this increment. Each company will have to fit its program to bring about the lowest final cost for its particular mode of operation.

High and Low Load Maintenance

In the Ohio Edison Company, we are continually studying our economy loading program and the various factors of which it is composed. We are aware of certain conditions which incur maintenance; among those are extreme high loads and very low loads.

Some of the conditions which create extra maintenance when equipment is operated at high loads are: (1) Superheater tube erosion. (2) Slagging. (3) Overheated tubes. (4) Stoker grate deterioration. (6) Induced draft fan erosion.

Similarly, the following conditions are among those which create extra maintenance expense when equipment is operated at low loads: (1) Air heater corrosion. (2) Mechanical dust collector stoppage. (3) Turbine blade erosion. (4) Condenser tube erosion. (5) Valve erosion (feedwater, boiler feedpump).

Several years ago, studies were made of the maintenance expenses of typical units which had operated under high load, or low load, prior to the scheduled maintenance outage so as to isolate those maintenance costs resulting from such high or low load operation. Low load decrement and high load increment costs were determined. (See Fig. 2.) From these studies similar costs have been estimated for similar units and assigned accordingly. In all cases, conservative values were used so as not to overstate the high load and low load portion of incremental maintenance. These costs were first calculated in terms of dollars per megawatt and were then converted to equivalent Btu/kwhr through the prevailing cost per million Btu of fuel. (See Fig. 3.)

$$\frac{\text{Increment Maintenance Cost}}{\text{Fuel Cost}} = \text{Equivalent Incremental Heat Rate}$$

or

$$\frac{\$/\text{Mw} \times 100 \times 10^6}{1000 \times ¢/\text{MM, Btu}} = \text{Equivalent Incremental Heat Rate, Btu/kwhr}$$

The incremental high load equivalent heat rate was added to the high load increment and the low load decrement subtracted from the low load increment when designing the heat rate cards which are contained in the station simulator sections of our GEDA economic dispatch computer.

These corrections increase the slope of the increment curve as the load increases. In the economic dispatch of the system this has the effect of increasing the load from the minimum values sooner and also prevents the units from remaining at the extreme high loads for long periods of time. In our particular application, it results in a broader dispatch of load changes by minimizing the effect of flat spots in the increment cost curves which

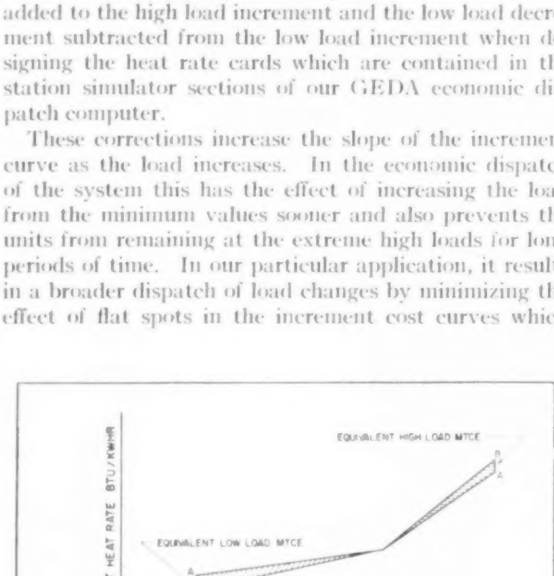

Fig. 3—Equivalent correction to increment heat rate data resulting from high and low load maintenance

otherwise would require one unit to take the brunt of the load changes at certain system load levels. Thus the regulation rate of response of the system is improved because more units are participating in the loading program at each system load level.

Incremental Coal Handling Maintenance

There are other maintenance charges which must be considered which can more accurately be described as a function of the amount of coal burned. Among these charges are maintenance of pulverizers, coal crushers, bulldozers, unloading cranes, ash handling equipment, and coal conveyors, and these costs can be readily determined. Fig. 4 relates the coal handling maintenance to high and low load maintenance and gives the net effective increment maintenance. Curve C-C is the increment of coal handling maintenance, which when corrected for the low load decrement and high load increment becomes Curve A-A. In our program, the coal handling maintenance is expressed in terms of cost per ton of coal consumed and is introduced into each generating unit's increment cost by the fuel cost dial settings on the GEDA computer.

Determination of Adjusted Fuel Costs for GEDA Computer

Each month an analysis is made of the current coal costs, heating values, and performance of the generating equipment. These data are combined with the incremental maintenance charges which can be stated as a function of amount of coal burned to obtain an effective fuel cost to be set into the GEDA Economic Dispatch Computer for each station on the system. Table II is a tabulation of typical data involved in this computation.

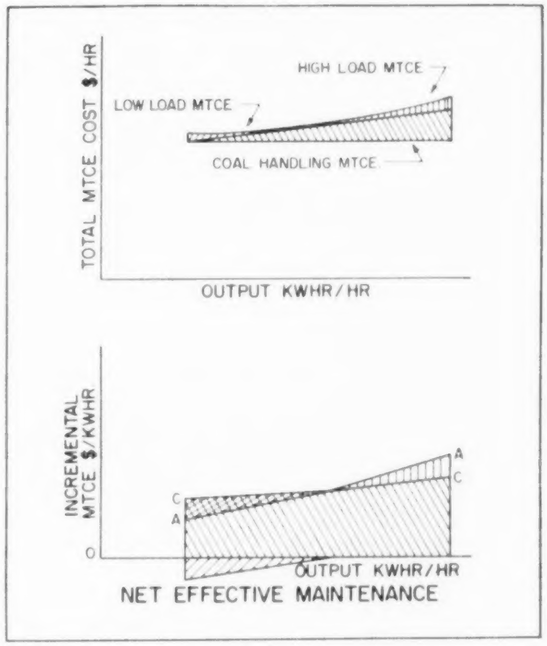

Fig. 4—Relationships of various maintenance costs and their equivalent incremental values

Conclusions

(1) In modern large generating stations, maintenance expenses now represent a greater portion of the total operating expenses excluding fuel. This increase makes the evaluation of incremental maintenance more significant.

(2) Detailed studies of those maintenance expenses which are known to be incremental in particular plants provide practical methods of evaluation. This can be done by analyzing separately: (*a*) High Load Main-

TABLE II ADJUSTED FUEL COSTS FOR GEDA (TYPICAL DATA)

Plant and Cycle		(a) Coal Burned, Btu/lb*	(b) Delvd. Coal Cost, $/Ton†	(c) Est. Delvd. Cost, ¢/MM Btu†	(d) Inc. Oper. Hdlg., $/Ton	(e) Inc. Mtce. Hdlg., $/Ton	(f) Perf. Factor, %*	(g) Adjusted Fuel Cost, ¢/MM Btu
Plant "A"	RH	11,922	4.20	17.61	0.07	0.15	100.23	18.49
	HP	11,974	4.20	17.54	0.07	0.15	99.68	18.52
Plant "B"	RH	11,688	6.67	28.53	0.04	0.15	100.00	29.35
	HP	11,539	6.67	28.90	0.04	0.10	94.26	31.31
Sta. Avg.		11,661	6.67	28.57	0.04	0.14	99.38	29.56‡
Plant "C"	HP	12,782	7.02	27.46	0.03	0.15	100.2	28.11
	LP	12,319	7.29	29.59	0.03	0.25	100.1	30.69
Sta. Avg.		12,650	7.08	27.77	0.03	0.16	100.2	28.49‡
Plant "D"	RH	12,244	5.17	21.11	0.03	0.15	95.53	22.87
Plant "E"	HP	11,965	4.09	17.09	0.04	0.15	97.3	18.38
	LP	12,169	4.09	16.80	0.04	0.25	91.4	19.70

* Previous month's actual.
† Next month's orders.
‡ Weighted on basis of estimated coal to be burned for each cycle.

The adjusted fuel cost (Column "g") is the sum of the delivered cost per ton, incremental operations handling cost per ton, and incremental maintenance handling cost per ton, such sum divided by the heating value and the performance ratio.

$$\frac{(\$4.20 + \$0.07 + \$0.15) \times 100 \times 10^6}{2000\ \text{lb/Ton} \times 11{,}922\ \text{Btu/lb} \times 1.0023} = \begin{matrix}18.49¢/\text{MM Btu}\\ (\text{Column “g”})\end{matrix}$$

It should be noted that the performance factor is essentially the ratio of the theoretical hourly inputs (from which the increment heat rate in the computer has been derived) to the actual experienced hourly inputs during a month's operation. However, this is a subject which could be covered in a separate paper and space does not permit elaboration on these details at this time.

tenance Increment; (*b*) Low Load Maintenance Decrement; (*c*) Maintenance Increment as a Function of Coal Consumption

Acknowledgments

The authors wish to acknowledge contributions of H. R. Roser, Ohio Edison Company, C. D. Zimmerman (Retired), Ohio Edison Company, and F. B. Pyle, now of Beiswenger Hoch and Associates, Akron, Ohio, for the development of much of the background material.

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Unique Water Table Solves Duct Design Problems

THE Buell Engineering Company, Inc., has devised a unique "water table" test procedure for studying the effects of inlet and outlet duct arrangements upon air flow patterns within electric precipitators which it manufactures.

The efficiency of these giant dust collectors is affected by the distribution and smoothness of the gas flow through the collector and hence it is desirable to design ductwork, vanes and baffles for uniform distribution and minimum turbulence. Each installation is different in that ducts leading to and from the collector are designed to meet particular space situations and seldom are any two alike.

Buell first began utilizing its water table system as an aid to achieving greater effectiveness from collectors in the field. The success of the system soon led to its adaptation for design purposes and now all preliminary ductwork design is tested on the water table before final design work is undertaken. In this connection, 80 per cent of the ductwork to and from collectors is laid out by the customer or his consultant, Buell reserving the right of final approval since the gas flow pattern has such a marked effect on precipitator efficiency.

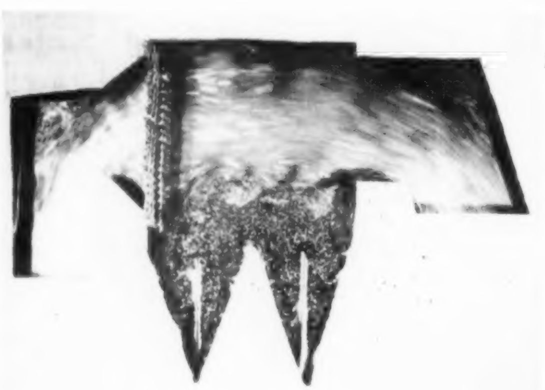
Before adapting the water table to general use, tests were conducted correlating water flow patterns with air flow patterns obtained in an actual duct. Results were sufficiently close to indicate that the water table data were reliable and subsequent field results have verified this contention.

The water table consists of a 3-ft × 5-ft × 5-in. deep table with a water inlet at one end and an outlet at the other. Discharged water is collected in a tank located under the table and is recirculated by means of a $^3/_4$-in. hp pump. Velocity is controlled by a valve in the 1-in. inlet pipe and this valve is generally fully open. Changes in velocity do not greatly affect the flow pattern but sometimes it is desirable to study the pattern in "slow motion." Strips of metal, 2 to 3 in. high, are arranged on the table on edge to form a cross-section of the precipitator, to scale of about $^1/_2$ or 1 in. to the foot. The cross section generally represents a vertical plane parallel to the flow through the precipitator. Horizontal distribution is generally sufficiently uniform, but vertical flow can extend into collector hoppers, set up eddy currents, and travel in strata at different speeds.

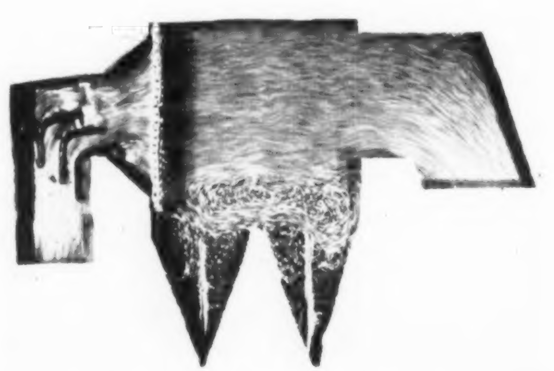
By trial and error, laboratory personnel have devised ways of observing the flow of water. The best method found to date is to dye the water blue with ordinary bluing and then sprinkle it at the inlet with aluminum powder. This creates an effective pattern indicating flow characteristics.

To extend the knowledge gained in the laboratory to the Buell engineering and service force and, more importantly, to make this information available to precipitator users, motion pictures are taken of the various arrangements tried in solving a particular problem. Also, time exposures are taken to indicate results of various arrangements. The flow patterns used to illustrate this article are time exposures, figures at bottom of page.

This technique is considerably more economical than trial and error tests in the field or even three-dimensional laboratory tests.



Baffles in the path of the gases improve flow pattern



Combined baffles and vanes give still better results.

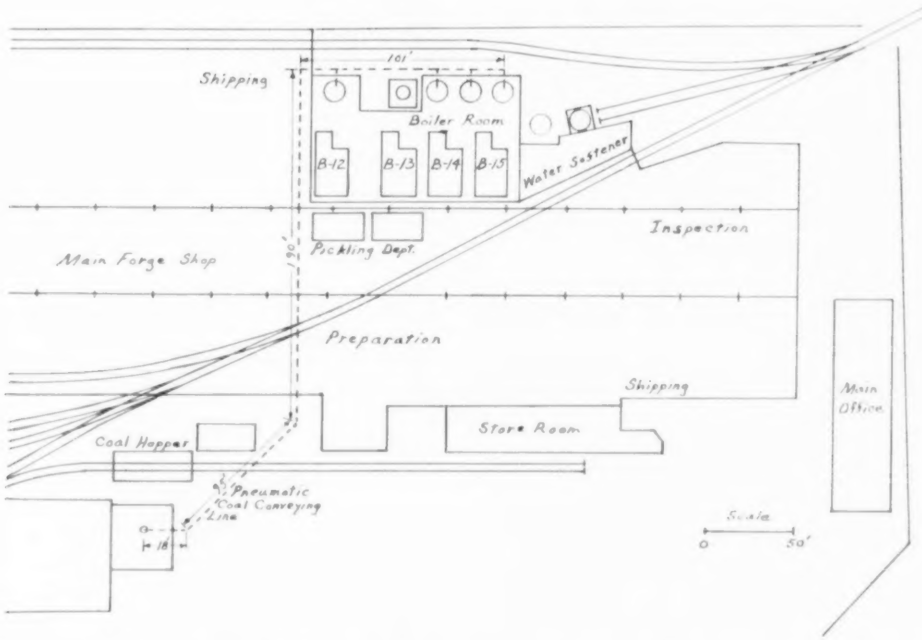


Fig. 1 Coal car hopper lies 450 ft. from the boiler plant. Railroad tracks, overhead cranes, other structures lie between so that a simple conveying system could not be designed. A pipeline, however, lent itself to such a task

Design Considerations for Pneumatic Coal-Handling System*

By WALTER GRUCA

Standard Forgings Corporation†

A 30-ton-per-hour-capacity pneumatic coal conveyer is described. Operating experience is discussed and design considerations are offered to eliminate difficulties experienced with this method of handling coal.

THE author's company is a producer of miscellaneous steel forgings for railroad equipment, farm and road machinery manufacturers, and other industries utilizing this type of material. Most of the forgings are produced on steam-operated hammers, requiring an aggregate steam producing capacity of 240,000 lb per hr. Over the years, the steam plant grew from an original installation of two boilers to a total of 29 boilers, of various capacities and relatively small size. Originally, the boilers were hand-fired.

In 1919, a major improvement in operating cost was realized with the installation of a central coal-pulverizing plant and conversion of the boilers from hand-firing to pulverized-coal-firing. This installation was still in use in 1937 when the first of the now existing 60,000 lb per hr boilers was installed, permitting the retirement of a number of smaller, less efficient boilers. Further additions to the boiler plant since then have resulted in the replacement of the old boilers with today's four 60,000 lb per hr units. The net result is a considerably more efficient boiler plant with enough capacity for the immediate future. The first three large boilers were installed dependent on the central pulverizing plant for fuel supply.

Changes in Pulverizing System

In 1953, when plans were made for the latest boiler addition, a study was made into the economic feasibility of converting the fuel supply system to the boilers from the central pulverizing plant to unit pulverizers at the boilers. This study indicated that savings in fuel and labor costs justified the conversion and it was decided to install fuel storage and processing at the boiler plant. Other indirect benefits realized by this change were the elimination of a fire and explosion hazard resulting from

* Contributed by the Fuels Division of the ASME for presentation at the ASME-AIME Joint Solid Fuels Conference, Old Point Comfort, Va., Oct. 9–10, 1958 as Paper No. 58-Fu 2.

† Works Engineer.

Fig. 2 The elevation view of Fig. 1 shows pneumatic pipeline runs 290 ft. underground, rises 25 ft. to another horizontal run of 100 ft. in the boiler house

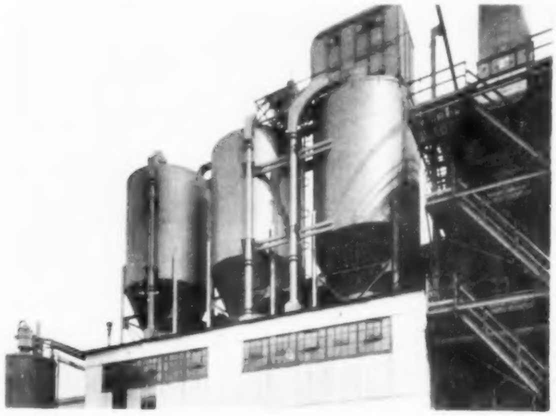


Fig. 3 Three of the receiving coal hoppers in Fig. 2

handling large quantities of pulverized coal and a source of air pollution. A major saving resulted from our ability to now use a lower grade and cheaper fuel.

The decision to install coal-handling and processing at the boiler plant brought with it the problem of coal supply to the boiler plant. The central coal-pulverizing installation was designed to receive coal in hopper cars and was well located with easy access to coal storage and supply. Trackage to hold a maximum number of cars was available in a location that caused no inconvenience to plant operation. The boiler plant on the other hand was situated in a corner of the plant area with limited trackage which could be reached only by moving cars out of company area and over a heavily traveled public highway. Also, with the limited trackage, moving and spotting cars for dumping would involve interference with plant shipping. In view of these considerations, it was decided to use the old car-dumping hopper at the central pulverizing plant and convey the coal to the boiler plant.

Study of Conveying Methods

This then led to a consideration of conveying methods. A conventional installation consisting of belts, elevators and other mechanical equipment was estimated at $115,000 because of the distance and elevation involved. When the suggestion was made to convey pneumatically, we were receptive to the idea because of the good experience we had with our then existing pneumatic coal conveyor. The central coal pulverizing plant distribution system consisted of a 4-in. pipeline using compressed air as the conveying medium which enabled us to move coal up to a maximum distance of 800 ft with very little difficulty. We realized that a great difference might exist between pneumatic conveying of raw coal and pulverized coal. But, after inspecting a few small raw-coal pneumatic conveyors and discussing operating experience with their operators, we decided that pneumatic conveying of raw coal was practical, especially after we compared installed costs. Our pneumatic conveyor cost $45,000, considerably under the cost of a mechanical system.

Equipment Layout

Fig. 1 shows the physical layout of the equipment and Fig. 2 is an elevation. As can be seen from the drawings the boiler plant is located approximately 450 ft. from the car hopper with railroad trackage, overhead cranes, and other structures between. Designing a simple conveying system around all of these obstructions was out of question. Installing a pipeline was a different matter. It was easy to plan a path that avoided interference and was another factor in our choice of the penumatic conveyor.

This conveyor is a pressurized type in that the conveyed material is fed into the conveying system operating at a pressure higher than atmospheric. 6500 cfm or 29,250 lb of conveying air per hr is furnished by a 200-hp positive-displacement rotary blower. Operating air pressure varies from a minimum of 1.5-psi with no coal in the system to a maximum of 6.5 psi when conveying to the most distant hopper. Coal is discharged into the air stream by means of an 18-in. diam, 22-in. wide pocketed rotor with six pockets rotating in a closely fitting housing. The pipeline is 12 in. diam with straight lenghts composed of 0.76-in. wall cast-iron pipe and bends made of 12-in. 0.50-in. wall standard steel pipe. Pipe lengths are joined with Dresser couplings except a few at valves or tanks which are flanged. Steel pipe bends are made to a 6 ft radius. All bends at which erosion takes place owing to change of direction are backed with concrete cast inside of a sheet-metal box welded to the external radius of the bend.

The overall length of the pipeline is 469 ft from the entry point to discharge at the farthest hopper. The first 291 ft are underground, crossing under two railroad tracks and two mill-type buildings where various manufacturing operations are taking place. The pipe rises vertically 25 ft out of the ground at a third building and then bends to the horizontal for a run of 101 ft in the boiler building. On this horizontal run are located three switching or diverter valves by means of which coal flow is directed into any of four coal-storage hoppers. The branching leads from this line elevate the coal to the top of the hoppers, a distance of 41 ft finally to discharge into the coal hoppers. From entry point to last hopper, the coal travels a horizontal distance of 393 ft and rises 76 ft.

A Dust-Tight System

With a conveying system of this nature, the coal hoppers are necessarily dust-tight with no openings to atmosphere except through a dust collector. All of the hoppers are interconnected to each other by 17-in. venting connections with one hopper receiving all of the vented air and discharging it through a three-compartment bag-type dust collector. To assist air venting and prevent excessive build up of air pressure in the coal hoppers a 15-hp exhauster draws on the dust collector and dis-

charges conveyor air to atmosphere. The three-compartment dust collector is continuously self-cleaning; during conveying, one compartment at a time is dampered off from the exhauster, a back flow of air is established through a by-pass damper, and the bags are shaken by a bag-shaking mechanism provided for each compartment. An electrical timer controls the proper sequence of operation.

Operating Sequence

At the entry point, coal is elevated from beneath the car hopper by belt conveyor to a roll ring crusher which reduces coal down to maximum $^3/_4$ in. size. The pneumatic conveyor was designed to handle this size of coal. Beneath the crusher and above the rotary feeder a small surge hopper directs the coal to the rotary feeder and smooths out variations in coal flow from the conveyor belt. Feed to the belt is by a reciprocating feeder from the car hopper, and amount of feed is regulated by a variable-speed drive on the reciprocating feeder.

Electrical interlocking is provided so that the equipment is operated in proper sequence and the accidental stoppage of any portion of the conveying circuit will shut down equipment ahead of it. The sequence of operation is as follows: the dust collector and exhauster are started first, next the conveyor blower, then the rotary feeder followed by crusher, conveyor belt, and finally the reciprocating feeder. A further safety measure is a pressure-sensitive switch to shut down the conveyor blower and coal feed if system pressure becomes too high. In addition, switching controls and indicator lights at this control point enable the operator to gage the amount of coal in each bin and direct coal to where needed.

The accompanying illustrations show various parts of the system. Fig. 3 shows the battery of three hoppers located adjacent to each other with the fourth hopper out of the photograph at a point 65 ft to the right. The riser pipes can be seen coming up through the boiler-house roof and extending to the top of the coal bins. The wear-resisting backing on the bends is plainly apparent. The dust collector on the right-hand hopper is also in view. Fig. 4 shows a pair of diverter valves and the pipe bend turning upward is the bottom of one of the risers in Fig. 3. The diverter valves are motorized and controlled remotely. Fig. 5 is that of the positive-displacement blower with conveyor control panel in the background. Fig. 6 is a view of the crusher and feeder pit, the entry point of the pneumatic conveyor. The conveyor pipe at this point is 10 ft below grade. Fig. 7 shows the control panel. The rotary switches are used to control diverter valves and by means of lights indicate when the valves are in position. Lights located in the hopper outlines on the panel board indicate low or high coal level in the individual hoppers.

Power to convey coal varies between 60 and 96 kw. The conveyor uses 60 kw to convey coal to the closest hopper and 96 kw to the farthest. Power consumption varies from 2.0 to 3.2 kwhr per ton, depending on which hopper is being filled. Conveyor air pressure for the shortest distance is 4 psi and 6.5 psi on the longest when conveying coal. Air flow is constant since air is supplied by a constant-speed positive-displacement blower except for variations caused by slip as pressure varies. Air flow is calculated to be 488 lb per min. At this rate 0.488 lb of air is needed to convey 1 lb of coal.

Performance

This conveyor system has proved reliable. It has delivered over 81,000 tons of coal in its 4 years of operation and no boiler outages have been caused by it. The basic design called for conveying a day's supply, up to 150 tons daily, to the boiler plant in an 8 hr period of time with essentially one-man operation. With dry and free flowing coal, one man can move cars over the hopper with a car spotter, empty the cars, and convey the coal to point of use without any assistance. However, during freezing weather with wet or frozen coal, we have found it necessary to assign a laborer to assist this man to empty cars, break up large frozen lumps of coal so that they will drop through the track hopper screen and to clean up under our conveyor belt where sticky fines falling from the conveyor belt tend to accumulate. Wet coal

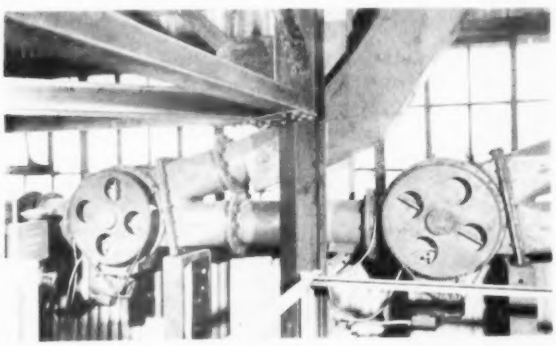


Fig. 4 Switching or diverter valves, such as the above, can direct the coal into any of the four coal hoppers, Figs. 2, 3

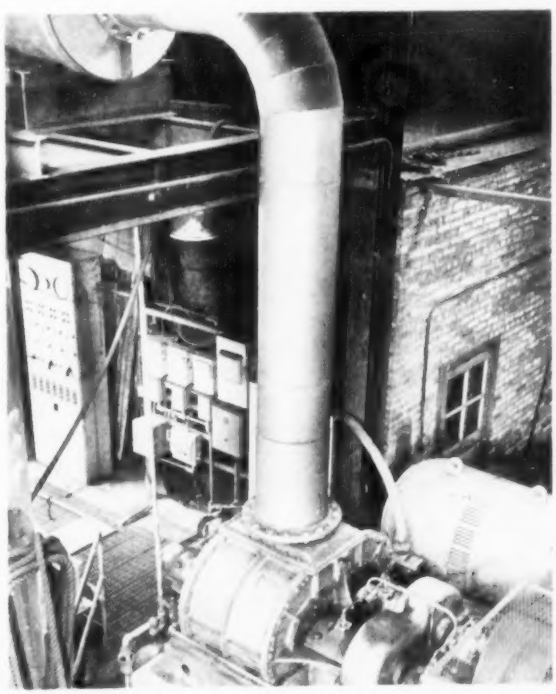


Fig. 5—Positive-displacement blower—conveyor control panel in background

Fig. 6—View of crusher and feeder pit which is the entry point for the pneumatic conveyor, Fig. 1

and freezing temperatures also cause us difficulties at our entry point. The pockets in the rotary-feeder rotor tend to fill with damp, sticky fines, which in turn reduces the volumetric capacity of the feeder. Under these conditions, either the volume of material passed to the feeder must be reduced or the conveying must be stopped and the feeder cleaned out by hand. This abnormal condition is usually preceded by a prolonged period of wet or snowy weather followed by freezing temperatures. Coal saturated with surface moisture and conveyor operation in ambient temperatures of 15 to 25 F seems to be the condition causing this difficulty. Above or below this range we have little or no difficulty. Fortunately, there are not too many days of this sort in our vicinity. In an effort to correct this condition which seems to be caused by fines freezing to the feeder rotor, we have provided a hollowed design of rotor into which we plan to inject steam in order to heat the rotor. This rotor was installed last winter but we had no opportunity to try it. While we had considerable difficulty with this condition in the winter of 1956–1957, last winter, 1957–1958, we had very little. By the time we completed installing the steam line, the necessity for it disappeared.

Rotary Feeder

As mentioned above, the rotary feeder consists of a pocketed rotor inside of a closely fitted housing. When new, clearance of the order of 0.005 to 0.010 in. is provided between rotor and housing. As wear progresses, this clearance increases and conveying air begins to leak through, interfering with the proper feed of coal. Takeup is provided, allowing the rotor to be adjusted closer to the housing. However, we found that with the cast-iron material in rotor and housing and uneven wear on both of these parts, 4 to 6 months use was the normal life of these parts. We have since salvaged the worn housings by flame spraying the worn surfaces with stainless steel and welding hardfacing material to the rotors. The first of these salvaged units is now in operation over a year and seems to be good for a longer term of service. In a new installation, we recommend that these parts be made of wear-resistant material.

Pipeline

The conveying pipeline will have a long life. We originally estimated a life of 10 years but our experience to date indicates a longer life. The straight runs of cast-iron pipe show very little wear. The steel pipe bends with the concrete backing are also holding up very well. The steel of the pipe at points exposed to erosion has disappeared but the concrete is resisting erosion very well. We have had wear beyond the concrete at two of the bends but a small weld repair quickly took care of this.

Diverter Valves

The diverter valves are another source of maintenance. As designed, these valves consist of a round cylindrical housing with sides enclosed by bolted flat plates. These plates carry the bearings which support the diverter gate. The gate is very closely fitted between the plates and the interior diameter of the cylindrical housing. Corrosion on the plates and buildup of a thin layer of damp coal fines on the plates and interior diameter of the housing causes the diverter gate to stick to the point that the drive motor cannot move it. When this happens, our only recourse is to dismantle the valve, clean out the fines and reassemble. During abnormally wet weather we have had to dismantle at least one valve a week. The diverter design is such that the gate must fit tightly to the side plates and housing to prevent leakage of coal and air, and since these surfaces are exposed to the stream of coal, corrosion and coating with coal fines naturally takes place. Our solution to this problem has been to redesign the gate so that it completely encloses the coal and air stream. With this design it is necessary to have a close fit only at the point of contact between the gate and the cylindrical body and no contact between the gate and side plates. These can now be made with sufficient clearance so that corrosion or a layer of coal fines will not cause binding. Unfortunately, we have no practical experience with this design having only recently ordered the necessary material to make the change.

Dust Collector

Since a pneumatic conveyor uses a considerable amount of air which must be vented, and since this air can carry a large quantity of dust, it follows that some form of dust collector is necessary at the venting point. In our case, the choice lay between a relatively inexpensive centrifugal dust collector or a considerably more expensive cloth-bag type of dust collector. Owing to the stringency of the air-pollution abatement ordnance in our locality we decided upon a cloth-bag dust collector. This type of collector does an excellent job in separating the dust from the air. However, in an area where local conditions permit, we believe a centrifugal collector would be more practical and less expensive to maintain. One unexpected condition we ran into with this conveyor is the

amount of moisture expelled with the venting air. While this is an excellent condition in predrying the coal and thereby improving pulverizer operation and increasing boiler efficiency, it did cause increased maintenance on the dust collector. The collector bags, particularly in the winter are always soggy. This did not prevent the free passage of air but did shorten the effective life of the collector bags. The best life we can expect from the cotton bags we use is two years. We had in mind to try other more expensive materials such as Orlon or Nylon but so far have not done so. Owing to this soggy condition which increases the weight of the bags, it is important that the bag supporting and shaking structure be more substantial than ordinary design calls for. This type of collector is usually designed for dry gas. We have had bag-support hooks break and supporting shafts bend from this condition.

Reliable Operation

Plugging of the conveyor pipe because of excessive moisture never happened with our conveyor. We have conveyed considerable amounts of snow and ice at times as well as soaking wet coal but it seems that once this material is dropped into the air stream it causes no difficulty at all. We have had power outages during the conveying cycle and experienced no difficulty in clearing out the line and resuming operation once the power supply was restored. Only on two occasions did we plug the line to the point that the conveying blower stalled. Each time the cause was traced to improper reassembly of the diverter valves. Limit switches on these valves which stop the gates in the proper position were misplaced, causing a partial discharge of coal and air into a branch which was supposed to be blocked off. Since sufficient air for proper conveying was not available, the coal was dropped into the pipeline until the line was full. When an attempt was made to use this line, it was not possible to operate the blower. On the first occasion, not wishing to overload the blower motor, we partially opened the line by hand but on the second occasion we took the risk and found that the blower was capable of clearing the line without damaging the motor. During these blockages, it occurred to us that a means for declutching the blower from the motor would be in order. It would then be possible to bring the motor up to speed before applying load. If a stream of air can be established through the blockage, the system can be cleared without any difficulty but with our rather large positive-displacement blower across-the-line starting, a small stream of air results in a serious overload. If provision is made to operate at a slow speed through a slip clutch this overload will not occur. One item which we did not provide, but which we believe is necessary, is quick-opening access points located at suitable spots on the conveyor line. Such locations would be at entry point, at bends, and at diversion points. This would simplify both inspection and maintenance at these points.

Pulverizer Requirements

Coal size degradation is not important to us since the final product is pulverized but where this is important, we would not recommend this type of conveyor. In our case, the amounts of fines and small sized coal at times caused us some difficulty with pulverizer operation, particularly with the pulverizer receiving its coal from the

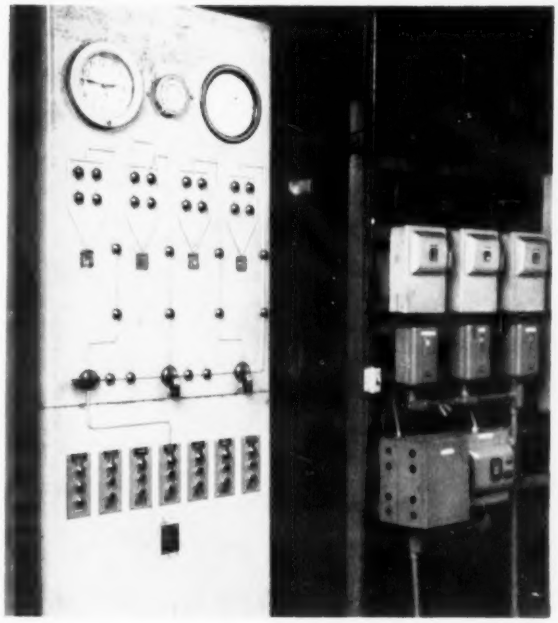


Fig. 7 Pneumatic conveyer control center carries a picture story of rotary switch operation for each diverter valve plus coal levels in hoppers

air venting hopper. This hopper tends to receive an excessive amount of fine dust which in turn causes pulverizer flooding by uncontrolled feed of fine coal. This condition is most acute when the pulverizer is operated after a prolonged shutdown and the hopper has accumulated a large quantity of fines. However, by installing a curved cut-off plate over the pulverizer rotary feeder, we seem to have corrected this condition judging by the fact that no further flooding was reported. Individual venting dust collectors on each hopper would have prevented this condition but for us it would have been an expensive solution with the type of dust collector we selected. If the less costly centrifugal collectors are permissible, we recommend them on individual hoppers to prevent the excessive accumulation of fines.

Absence of Hazards

Fire and explosion hazard with this conveying system seems to be nil. At least, in the four years of operation we have not had any difficulty on this score. We made every effort to eliminate tramp iron at the entry point by providing a magnet over the feeding conveyor belt, a magnetic head pulley, and our choice of roll ring crusher was guided by the fact that this type of crusher screens material passing through it and rejects any material which will not pass through the screen.

Because we designed the system for one man operation and in view of the distances and elevations involved, gaging coal level accurately in the hoppers was important to us. Diaphragm-type material level-sensing devices did not operate well on our system owing to the rapid variations of air pressures in the tanks. This problem was solved by using electronic level indicators which are not affected by this condition. However, these are still subject to build-up of damp fines on the sensing probes and must be regularly inspected and cleaned.

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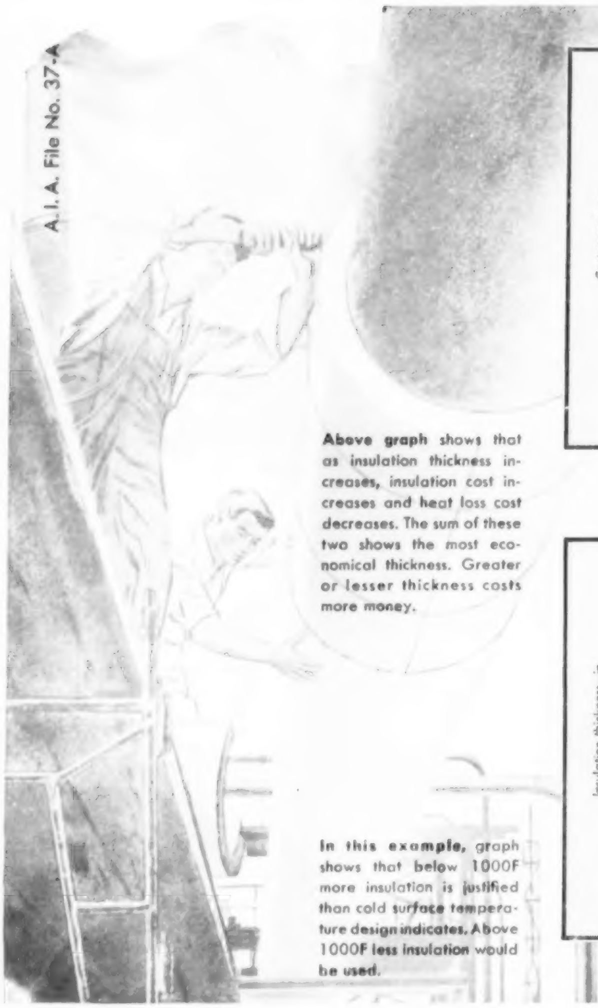
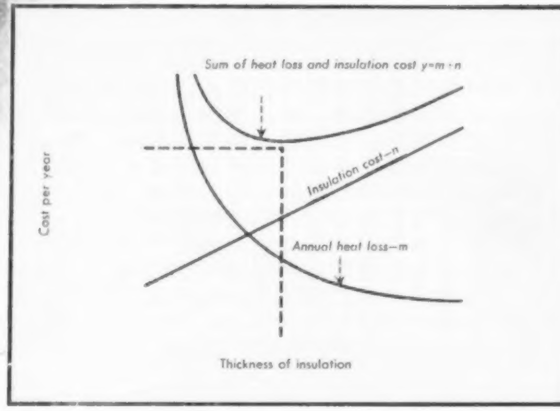
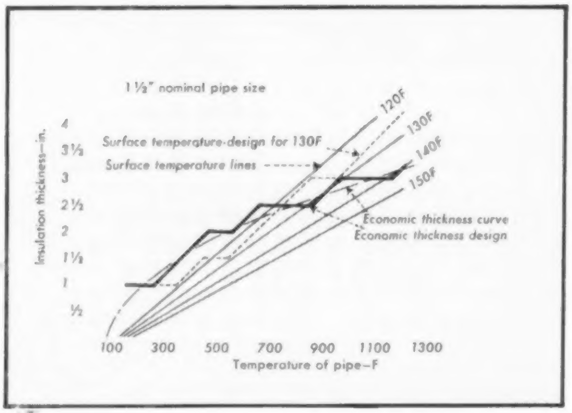




Above graph shows that as insulation thickness increases, insulation cost increases and heat loss cost decreases. The sum of these two shows the most economical thickness. Greater or lesser thickness costs more money.



In this example, graph shows that below 1000F more insulation is justified than cold surface temperature design indicates. Above 1000F less insulation would be used.

"COLD SURFACE" CRITERION WASTES MONEY!

How J-M engineers determine economic insulation thickness

to give you more for your insulation dollar

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Here's how it works. The annual cost of the heat loss through the insulation is plotted for various thicknesses. Also plotted is the annual cost of insulation. A third curve is then drawn as the sum of heat loss and insulation cost. The economical thickness is found where this third curve reaches its lowest point.

To arrive at the above figures in a given instance requires the following: 1. Cost of heat production per million Btu; 2. Rate of heat loss through insulation in Btu per unit area per hr; 3. Annual hours of operation; 4. Applied cost of insulation per unit area; 5. Rate of amortization and required return on the insulation investment cost. Items 2 and 4 are available from the insulation manufacturer; others are normally supplied by the plant engineer.

For more complete information, call or write Johns-Manville, Box 14, New York 16, New York. In Canada, Port Credit, Ontario. Ask for reprint of technical article "Select Economic Insulation Thickness."

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Abstracts From the Technical Press—Abroad and Domestic

(Drawn from the Monthly Technical Bulletin, International Combustion, Ltd., London, W. C. 1)

Fuels: Sources, Properties and Preparation

Science in the Use of Coal. R. L. Brown. *Nature* 1958, **182** (July 5) 24–6.

A brief account is given of the papers presented to the Conference on "Science in the Use of Coal" held at Sheffield in April 1958 by the Institute of Fuel. The various sections were concerned with: 1. Physics and chemistry of coal; 2. Preparation and breakage of coal; 3. Carbonization; 4. Combustion and gasification; 5. Reactivity of cokes and chars.

Pore Structure. Anon. *Coal Research* C.S.I.R.O. 1958, No. 4 (May) 4–7.

The methods used to measure the size and distribution of pores in coal and coke are described and the application of this knowledge to the investigation of combustion and carbonization processes is indicated.

Mode of Occurrence of Chlorine in Coal. G. N. Daybell and W. J. S. Pringle. *Fuel* 1958, **37** (July) 283–92.

A substantial part of the chlorine in coals from the central British coalfields is present as chlorine ions attached to the coal substance by a linkage with ion exchange properties. Calcium and magnesium chlorides which appear in water extracts of coal are shown to be formed by reaction between carbonate minerals present and hydrochloric acid produced from this ionic chlorine. The coals contained water-soluble sodium equivalent to from one-third to a half of the total chlorine content.

From *authors' abstract.*

Thermal Decomposition of Coal in High Vacuum. Bozen Sun, C. H. Ruof and H. C. Howard. *Fuel* 1958, **37** (July) 299–308.

Decomposition of four coals of different rank in an all-steel molecular distillation apparatus has shown that the distillate from the coal with the strongest agglutinating properties is richest in high molecular weight components. All the coals studied failed to pass through a fluid stage and to form a coherent coke under the experimental conditions. There appears little doubt that coke formation depends on the liberation of certain high molecular weight, but fusible components in the early stages of decomposition.

Optical Properties of Graphite and Coal. J. T. McCartney and S. Ergun. *Fuel* 1958, **37** (July) 272–82.

The optical constants of graphite and coals were determined and their variation with metamorphism studied. A relationship between log reflectivity and H/C ratio and a similar one between absorption index and H/C ratio were observed, but the variation in the refractive index with metamorphism was small.

The Infra-red Spectra of Coal and Some Coal Derivatives. W. den Hertog and N. Berkowitz. *Fuel* 1958, **37** (July) 253–71.

Desulferization of Coke-oven Gas at Appleby-Frodingham. L. Reeve. *J. Inst. Fuel* 1958, **31** (July) 319–24.

The pilot plant for treating 2.5 mil cu ft of gas in a fluidized bed of iron oxide at a temperature of 350 C is described. The sulfided oxide is regenerated in a current of air at 600–800 C and recycled.

Heat: Cycles and Transmission

Calculations of Heat Exchange in Combustion Chambers. P. K. Konakov, S. S. Filimonov and V. A. Khrustalev. *Teploenergetika* 1957 (Aug.) 48–53 (In Russian). (D.S.I.R. translation C.T.S. No. 499).

A method of calculating heat exchange in boiler furnaces is outlined.

Some Problems in the Theory of Heat Transfer in the Case of Laminar Flow of Liquids in Tubes. D. A. Labuntsov. *Teploenergetika* 1958, **5** (Mar.) 55–60 (In Russian).

Heat transfer in a hydrodynamically stabilized laminar flow is discussed. A general solution is advanced for the case where heat supply is steady, and which permits determination of the length of the zone of thermal stability. The influence of axial heat conductivity in the flow on heat transfer, when tube wall temperature and heat supply, respectively, are constant, is considered.

From *C.E.G.B. Digest* 1958, **10** (Aug. 2) 1986.

Experimental Investigation of the Mechanism of Heat Transfer with Surface Boiling of Water. G. G. Treshchov. *Teploenergetika* 1957, **4** (May) 44–8 (In Russian). (D.S.I.R. Translation C.T.S. No. 465, 1958).

Experimental work carried out to explain the mechanism of heat transfer with surface boiling of water and the high rate of heat transferred under these conditions is described. The results are presented in graphs and some equations fitting these graphs have been developed.

Steam Generation and Power Production

Pulsating Pressure Fatigue Tests on Pressure Vessel Branch Connections. P. H. R. Lane. *Brit. Weld. J.* 1958, **5** (July) 327–32.

Four different welding details (two interpenetrating and two not fully penetrating) used for attaching branches to 1 in. thick pressure vessel subjected to pulsating pressure were observed. The fractures showed that the root area of the not fully penetrating welds was not a source of weakness.

From *author's abstract.*

Film Boiling on Boiler Surfaces. E. Zielinski. *Combustion* 1958, **29** (June) 47–50.

The reasons for the occurrence of film boiling in boiler tubes have been investigated experimentally and it has been found that, if a tube surface after having been out of contact with water due to a steam plug or scale is wetted again, film boiling will occur even at low heat flux. It is argued that not the presence of scale but its breaking off and the sudden contact of water with the hot tube wall with consequent film boiling causes overheating and tube failures.

The Influence of Surface Roughness on the Flow Resistance in Narrow Pipes. G. Grass and E. Luth. *Chem. Ing. Tech.* 1958, **30** (July) 447–9 (In German).

The validity of Nikuradse's equation for the flow resistance in rough pipes was experimentally verified for pipes of 10 mm O.D. and 1 mm wall thickness, using water at a temperature of 13 C under a constant pressure of up to 2000 mm w.g. The influence of surface treatment to reduce roughness on flow resistance was also investigated and it was shown that the resistance decreases with increasing smoothness, especially at high flow velocities (20 m/s) whereby the resistance may be reduced by up to 18 per cent.

Choosing Draft Systems for Industrial Boilers. H. G. Meissner. *Power* 1958, **102** (July) 98–100, 151.

A comparison is made between the advantages and disadvantages of bal-

anced draft and pressurized furnace firing and the reasons are given why the latter is increasingly applied in the smaller industrial boiler. Pressurized furnace firing is applicable only to pulverized coal, gas and oil fired furnaces, but its extension to traveling grate and spreader stokers with continuous ash discharge is under development.

Optimum General Temperature for Regenerative Preheat of Feedwater. D. D. Kalafati. *Teploenergetika* 1958, **5** (Mar.) 60–7 (In Russian).

An analytical investigation is made of the optimum temperature for feedwater regenerative preheating in steam turbine sets, variation in efficiency of the boiler unit being taken into account. Consideration is given to the optimum general temperature in the case where intermediate steam preheating is used.

From *C.E.G.B. Digest* 1958, **10** (Aug. 2) 1958.

The State of Steam Generator Technique. H. Richter. *Mitt. V.G.B.* No. 54, 1958 (June) 197–210 (In German).

This second analysis of the data contained in the files of the V.G.B. evaluates according to type and size of boiler, steam parameters employed, and firing method: (1) Type of superheater (pendent or horizontal, tube platen and/or convection); (2) reheaters; (3) steam desuperheaters; (4) flue gas exit temperature; (5) I.D. fans.

High Pressure Metal Casings. II. Boiler Drums. J. Theodore. *Soudage* 1958, **12** (May/June) 211–6 (In French).

The steels used, drum dimensions and steam pressure, acceptance tests on plates, forming, welding, pre- and post-weld treatment, and inspection of welds are described.

Furnaces and Combustion

Development of the Pulverized Coal-fired Air Heater. K. Bammert. *VDI-Z.* 1958, **100** (July 11) 841–50 (In German).

The development of the coal-fired air heater for the 2 Mw closed cycle gas turbine plant at Ravensburg is described and details of design and construction are given. Operational experience has been very satisfactory; tube failure has not occurred and the air temperature could be held constant at 655 to 665 C between one-quarter and full load. The designs of a simple air heater for a 6.6 Mw plant and twin air heater for a 13.75 Mw plant are shown and the essential data tabulated.

Research Tasks and Development Tendencies in Furnace Design. W. Gumz. *Mitt. V.G.B.* No. 54, 1958 (June) 180–97 (In German).

A broad review is presented of: (1) Present designs of stoker and pulverized fuel-fired furnaces and burners, the ignition of low-volatile fuel, maximum temperatures in wet and dry bottom furnaces; (2) the influence of flow conditions in burners and furnaces on ignition and combustion; (3) the application of results of recent research in burner, furnace, and boiler designs; (4) further research and development tasks.

Combustion Processes and Particle Motion of Pulverized Coal. S. V. Bukhman. *Izvestiya Akademii Nauk Kazakhskoi S.S.R.* Power Industry Series 1956, No. 11, 70–81 (In Russian).

Results of calculations and test data on the motion of particles of unchanging mass and form confirm the possibility of determining such motion in a stream with considerable accuracy. In motion calculation, variation in mass of the particles may be disregarded to a first approximation, and they may be taken as spherical. Tests to determine surface rate of burning and the surface temperature of particles of electrode carbon confirm the theory. The possibility is indicated of direct measurement of the temperature of falling burning particles by the photographic pyrometer method.

From *C.E.G.B. Digest* 1958, **10** (Aug. 2) 1958.

Experimental Study of Radiation of Pulverized Coal Flame due to Fuel Burn-out. A. A. Demin. *Izvestiya Akademii Nauk Kazakhskoi S.S.R.* Power Industry Series 1956, No. 11, 82–96 (In Russian).

The relationship of radiation to the burning characteristics of a pulverized fuel flame, when combustion, radiation and temperature are measured simultaneously, is studied. A quantitative relationship between the blackness factor of the flame and the degree of fuel burn-out is obtained. It is established that increase in the excess-air factor is accompanied by decrease in the blackness factor of the flame up to a certain point, after which the latter factor remains fairly constant. The dependence of the blackness factor of the radiation on flame size is again confirmed.

From *C.E.G.B. Digest* 1958, **10** (Aug. 2) 1954.

Method and Apparatus for Heating Fluids. Oxy Catalyst Inc. *British Patent* 797,574 U.S.A., 25th July, 1955.

The system consists of two beds. The lower one contains a high activity catalyst through which air preheated to 400–600 F and a small quantity of fuel are passed to heat the air to 1000–1500 F and where the catalyst bed undergoes little movement to prevent attrition of the catalyst particles. The highly preheated air and the major part of the fuel are passed under fluidizing conditions through the upper bed of the low-activity catalyst (copper, silver, chromium) to obtain a temperature of 1000–1200 F and where the medium flowing through the tube bundles immersed in the bed is heated.

Water-Side Corrosion and Water Treatment

Principles and Experiences with Modern Feedwater Treatment. R. Freier. *Techn. Mitt.* 1958, **51** (June) 244–7 (In German).

A brief review is presented of feedwater specifications suitable for all types of boilers, the methods available for attaining these specifications, the modernization of older plants, and recent experiences with complete demineralization plant.

Calculation of the Economics of Complete Demineralization Plants. R.

Brunner. *Mitt. V.G.B.* No. 54, 1958 (June) 210–4 (In German).

Feeding of High-pressure and Super-critical-pressure Once-through Boilers with Chemically Desalted Water. N. V. Bulgakova, Z. V. Deeva, A. A. Kot and K. A. Rakov. *Elekt. Stantsii* 1958, **29** (Mar.) 8–12 (In Russian).

Tests were carried out on an experimental once-through boiler in 1956 to ascertain the behaviour of salts in the boiler on feeding with condensate and chemically desalting the mains water received. The salt content of water and steam was determined by ion exchange filters, and the silica content according to the blue complex of molybdenum silicate acid on a photoelectric calorimeter without extraction. The results of the five tests made are set out in a table, supported by six graphs.

From *C.E.G.B. Digest* 1958, **10** (July 19) 1855.

A New Development in Boiler Water Treatment. N. H. Peters. *Steam Engr.* 1958, **27** (July) 327–31.

The treatment developed in cooperation by I.C.I. Ltd. and Alfloc Water Treatment Service to prevent scale formation in boilers and to make the sludge free flowing, previously described by J. A. Gray, E. F. Thurston, and L. Furnival, is recounted and results of applications in a wide variety of boilers and with very different waters are added. A mobility rating for the sludge has been established (1 very poor, 9 very good) and a formula for calculating the flow characteristics of sludge is given. If the mobility rating is below 7, magnesium is added so that the magnesium hardness in the water is at least three times the silica plus one fifth of the calcium hardness. Treatment based on the formula has now been applied to 450 plants and, where the specified conditions were maintained continuously, scale formation was negligible (less than 0.01 in. thickness) and the sludge remained free flowing so that it could be readily removed with the boiler blowdown.

Corrosion and Water Treatment. G. G. Sindery. *Corrosion Tech.* 1958, **5** (July) 209–12, 224.

Certain water treatment methods actually increase the corrosiveness of the water, but this can be greatly reduced by correct selection of pretreatment and conditioning treatment. A number of corrosion inhibitors, especially organic inhibitors, are described.

Theory and Practice of Boiler Water Treatment as a Means of Eliminating Corrosion and Scale. E. Quaade Holm. *Corrosion Tech.* 1958, **5** (July) 213–4, 237.

Causes of corrosion in boilers, such

as the presence of atmospheric oxygen and differential aeration, are described and methods of feedwater treatment preventing corrosion and scale formation are outlined.

Magnetic Treatment of Liquids for Scale and Corrosion Prevention. T. Vermeiren. *Corrosion Tech.* 1958, **5** (July) 215–9.

The apparatus used in the magnetic treatment of liquids is described, by which the crystallization of substances contained in the water is promoted. The treatment does not eliminate hardness salts, but alters them physically and when this process is applied to boilers, a very fine mud is formed which can be removed by blowdown.

"Pulsometer-Tett" Water Treatment Processes. Anon. *Corrosion Tech.* 1958, **5** (July) 225–6.

The pulsometer filtering treatment of highly polluted and turbid water, using specially activated silica in conjunction with other chemicals, is described. For the removal of amoeba and algae, diatomaceous filters are used and for softening the water, ion exchange resins made of synthetic polymers.

Gas-Side Corrosion and Deposits

Investigation on a p.f. Boiler with an Output of 60/75 t/h to Establish the Causes of Excessive Fouling of External Heating Surfaces. J. Tyll. *Energetyka Przemyslowa-Gospodarka Cieplna* 1957, No. 1, 20–5, No. 2, 57–60 (In Polish).

For substratum layers of agglomerates formed on screens and superheaters the proportion of SiO_2 in samples of agglomerates ranges up to 50 per cent. The high proportion of silica is caused by the oxidation of silica and silicon sulphides reaching the boiler tubes. Care must be taken to mix fuel with air in correct proportions and to wet the air blast properly, since water vapour has an oxidizing effect on flue gases. Good results can be obtained by providing the furnace chamber with fixed equipment for blowing in desilting water from the boiler. Protection against agglomeration of deposits on the superheater constitutes a separate problem, which should be solved by so redesigning superheaters as to prevent excessive temperatures in tube walls, and by the smoothness of such walls.

From *C.E.G.B. Digest* 1958, **10** (July 19) 1830.

Tube Wastage in Slagging Furnaces. Evaluation of Operational Experiences. A. Bachmair. *Mitt. V.G.B.* No. 54, 1958 (June) 146–53 (In German).

Wastage of refractory lined tubes, gas-side erosion and corrosion and waterside corrosion are mainly caused by a reducing atmosphere which in its turn is due to an unsuitable burner design or position or incorrect fuel to air ratio. Contributory causes are failure to take into account the influence of the viscosity of the slag and its effect on the refractory lining, the flow conditions in the furnace, and the method used for reintroducing the fly ash into the furnace.

Tube Failures in Oil-fired Boilers. F. Glaubitz. *Mitt. V.G.B.* No. 54, 1958 (June) 156–60 (In German).

Tube failures caused by overheating in three oil and gas-fired boilers (two 80 klb/h, one 130 klb/h, 620 psi, 890 F) are reported, which occurred almost exclusively in the oil burner zone. Investigations showed that the heat release rate in the oil burner zone was about 90,000 Btu/sq ft, hr, in the gas burner zone about 57,000 Btu/sq ft, hr, and 10 ft above the oil burner zone 32,500 Btu/sq ft, hr, and that these high rates led to the deposition of scale containing 30 per cent Fe, 32 per cent SiO_2, 16 per cent Cu, 4 per cent Ca, and 1 per cent Na. This confirmed earlier Russian investigations that in boilers with high heat release rates copper-containing de-

posits must be expected if a certain limit is exceeded. It is hoped that by the installation of a complete demineralization plant for the makeup water and by passing the condensate through a filter to remove the Fe and through a cation exchanger and mixed-bed filter to remove the Cu, a further deposition of scale will be prevented.

Low Temperature Corrosion in Oil-fired Steam Generators. M. Lang. *Energie* 1958, **10** (June) 227–30 (In German).

A small traveling grate fired boiler (22,000 lb/h, 60 psi) was converted to oil firing and this resulted in rapid clogging and corrosion of the economizer, despite (or because of) regular soot blowing with steam. A change to soot blowing with compressed air (100 psi) once a week entirely prevented clogging and reduced the rate of corrosion. A larger boiler (76,000 lb/h, 60 psi) converted to oil firing with an addition of dolomite (0.08 weight per cent) was cleaned only with compressed air once a week and showed neither clogging nor corrosion. The reasons for the advantagesof soot blowing with air over steam are discussed.

Method for the Operation of Furnaces and Improvements In or Relating to Furnaces. British Petroleum Co. Ltd., R. D. Wilsdon and L. K. Rendle. *British Patent* 797,439, 13th March, 1956.

The patent related to the injection of ammonia (0.5 to 9 per cent by weight of the total weight of sulphur in the fuel) at or below 500 C into oil-fired furnaces to prevent low temperature corrosion.

Study of the Properties of Fly-ash Cement. M. Venuat. *Rev. des Mat. de Construction et de Trav. Publics* 1957, No. 506 (Nov.) 309–17 (In French).

Tests were carried out on two station ashes which were tested at different degrees of fineness. They have replaced artificial Portland cement generally in the proportions of 25 and 40 per cent. Sixteen mixes were tested and the strengths were found to be reduced only in the initial stages. The same was the case with the heat of hydration. Strengths are considerably increased by the action of heat. Positive conclusions are presented as regards improved pozzolanic properties due to milling. The ash has the advantage of reducing shrinkage even when finely milled.

From *C.E.G.B. Digest* 1958, **10** (July 19) 1848.

Plastic Models Improve Dust Collector Results. Anon. *Combustion* 1958, **29** (June) 41–5.

Model tests by Research-Cottrell have shown that considerable improvements in precipitator performance and reduction in fan power can be achieved by changes in the design of the ducts leading to the precipitator. The main object has been to reduce pressure loss and turbulence and this has been obtained by avoiding abrupt changes in cross-section and the introduction of perforated plates, where position and open area are both very important. Examples of the results obtained by these model tests are illustrated.

Heat Recovery Plant

Specific Heat Rate of Heat Exchanger Heated with Hot Steam, Especially of Feedwater Preheaters. S. Kriese. *BWK* 1958, **10** (July) 323–5 (In German).

Equations are derived for the specific heat rate and logarithmic temperature difference of heat exchangers in which the media change their phase. The equations are applicable without limitations and are independent of temperature conditions.

The Mean Temperature Difference in Multiple Heat Exchangers. H. Kühne. *Chem. Ing. Tech.* 1958, **30** (June) 404–6 (In German).

A new graphical method has been

worked out for calculating the mean temperature difference in heat exchangers in which counter, parallel, and cross-flow occur.

Damage to Wood in Cooling Towers by Chemicals and Organisms. H. Hannig. *Mitt. V.G.B.* No. 54, 1958 (June) 161–5 (In German).

Damage caused in cooling towers by chemicals contained in the cooling water and by algae and fungi is described and preventive measures by the impregnation of the wood with organic and inorganic chemicals are indicated.

Power Generation and Power Plant

Reheat Practice in British Power Stations. I. F. H. S. Brown and J. W. H. Dore. *Steam Engr.* 1958, **27** (July) 338–42.

The reasons for gradually increasing the size of units installed in power stations, the steam pressure and temperature, and for applying reheat in the latest units are set out and the problems introduced by these measures are discussed. The design of furnaces, superheaters and reheaters, their positions in the boiler and the control of superheated and reheated steam temperatures are considered.

Unit Generating Plant. H. Topley and G. Nicholson. *Elect. Rev.* 1958, **163** (July 11) 53–61.

Tests on a 100 Mw boiler-turbine unit involving starting from hot and cold, are described. Normal cold starts take about six hours, a hot start after six hours shut down about two hours. Observations are made on steam conditions at the turbine inlet valve and the maximum temperature rise.

Experience in Starting and Operation of High Pressure Unit Power Station. W. Nistler. *Techn. Mitt.* 1958, **51** (June) 237–43 (In German).

The procedures worked out for the starting of a Benson boiler and turbine from cold and after a short shut down and for the shut down of the unit are set out in detail. When starting from cold a maximum time of 90 min is required, but this can often be considerably reduced.

Some Possibilities of Improvements in the Steam Power Process. W. Bolte. *Energie* 1958, **10** (June) 223–7 (In German).

Two possibilities of increasing the thermal efficiency of power generation by improvements in the feedwater preheating circuit have recently been tried out. One is preheating above the saturated steam temperature corresponding to the steam pressure by desuperheating of the extraction steam, and the other is the final pre-

heating of the feedwater by flowing superheated extraction steam of a much lower pressure stage. Calculations show that both lead to a small but acceptable increase in thermal efficiency. Further possibilities are given by improvements in the stage distribution of preheating in reheating installations.

Big Town Power. M. L. Branin. *Coal Utilization* 1958, **12** (June) 18–22.

The Consolidated Edison Co. supplied in 1957 a 600 sq mile area of New York City and Westchester County with a population of 8.8 million with 15,400 mill. kwhr of electricity, 38,000 mill. cu ft of gas, and 16.8 mill. klb of steam, consuming 6 mill. tons of coal, 1.6 mill. coal equivalent tons of oil and 168.7 mill. cu ft/day of natural gas. There are nine electric power stations, three gas manufacturing plants, and six central steam plants. The total installed capacity in 1957 was 3717 Mw and the peak load of June 26 was 3460 Mw. The largest station at present is Hudson Gold in Brooklyn, but the Astoria station is planned for over 1000 Mw of which two 180 Mw units are in operation and a third unit of 335 Mw is to start in the autumn of 1958.

Materials and Manufacturing Processes

Pressure-welded Joints of Austenitic and Ferritic Steels (System Mannesmann). H. Linden and H. Henneke. *Mitt. V.G.B.* No. 54, 1958 (June) 165–75 (In German).

Mannesmann have developed two kinds of joints: one for tubes below $2^{3}/_{4}$ in. O.D. and one for those above. For the smaller sizes a ferritic tube clad with austenitic steel on the inside is used. For larger tubes, two jointing tubes are used which are given a conical shape and fit into each other so that the austenitic part is inside and which are then pressure-welded at a suitable temperature in a die, followed by heat treatment. The two parts may also be given a screw thread, though this has not proved essential. For use at a temperature above 500 C (1000 F) a thin nickel layer (30μ) is interposed to prevent carbon diffusion. A separation of the two steels has not occurred in a large number of tests carried out; in severe thermal shock tests, fine cracks have appeared on the surface of the austenitic steel, but this has in no way diminished the strength of the joint. Ultrasonic testing of the jointing piece is recommended.

Stresses in the Jointing Area of Two Tube Materials with Different Thermal Expansion Coefficients and Unequal Moduli of Elasticity. A. Schiller. *Mitt. V.G.B.* No. 54, 1958 (June) 176–80 (In German).

Eequations are given for calculating the stresses in joints of austenitic and ferritic steels. It is shown that these stresses are highest at room temperature and diminish with increasing operational temperatures. The stresses also decrease during operation as a function of time under the effect of creep.

Stress Corrosion and Structure Formation of Austenitic Chromium-Nickel Steel X8 Cr Ni Mo V Nb 16/13. E. Baerlecken and K. Lorenz. *Mitt. V.G.B.* No. 54, 1958 (June) 215–9 (In German).

Stress corrosion cracks in austenitic steel caused by accidental addition of sodium chloride to feedwater were investigated. It was shown that the steel from one source containing less nitrogen than that from another source was more liable to stress corrosion and that the Nb/C ratio was a secondary factor. In addition to metallographic examination, the corrosion behaviour of steels in fuming 65% nitric acid has proved a valuable indicator of the factors contributing to the crack formation.

Nuclear Energy

World Survey. Anon. *Nucl. Engng.* 1958, **3** (Feb.), 62–5.

Brief details are listed of nuclear development in sixty countries, the name and address of the State Authority and principal officials and reactors commissioned or under construction.

Technical Literature of 1957. Anon. *Nucl. Engng.* 1957, **3** (Feb.), 70–7.

The review is divided into the following headings: (1) Research reactors; (2) fast reactors; (3) trends in nuclear power; (4) power reactors; (5) power reactors, general; (6) graphite moderated, gas-cooled; (7) graphite moderated, sodium-cooled; (8) water moderated reactors; (9) beryllium moderated; (10) ship propulsion; (11) radiation hazards and personnel protection; (12) reactor control; (13) radiation shielding; (14) nuclear engineering in general; (15) books published.

Atomic Review. Continuous Process. Anon. *Engineering* 1958, **185** (Feb. 21), 236–8.

A summary is given of papers presented to the January 1958 meeting of the British Nuclear Energy Conference dealing with disposal of fission wastes, hazard of inadvertent criticality in chemical plants and aqueous homogeneous reactors.

Corrosion in Nuclear Power Production. B. L. Harbourne. *Corrosion Prev. and Contr.* 1958, **5** (Jan.), 43–7.

Corrosion of canning materials used in various types of reactors is discussed. For gas-cooled reactors "Magnox C" has proved its corrosion resistance in laboratory tests with wet and dry CO_2 and has been used for the fuel elements at Calder Hall. For the sodium-cooled reactor it is essential that the canning material is free of impurities since these accelerate corrosion. For the pressurized water reactor zirconium and some of its alloys and stainless steels are the only materials known so far to have sufficient corrosion resistance.

Fuel Elements for Use in Nuclear Reactors. United Kingdom Atomic Energy Authority and I. H. Morrison. *British Patent* 790, 389, 14th October, 1955.

The fuel element is provided with a series of circumferential grooves of rectangular section into which the sheath, preferably magnesium, is indented.

Design of Six Boilers for Gas-Cooled Nuclear Power Reactors. P. S. Otten. *Power* 1958, **102** (Feb.), 80–3.

The following designs are described and their advantages and disadvantages discussed: (1) Cross-flow steam generator; (2) a shell and tube steam generator; (3) bayonet-tube heat exchanger; (4) U-shell heat exchanger; (5) kettle-type steam generator; (6) vertical head-down bayonet heat exchanger.

Argonne Reactor Gives 1250 kW Dividend. Anon. *Elect. World* 1958, **149** (Jan. 13), 47.

At the Argonne Experimental Boiling Water Reactor (EBWR) it has been found possible to increase heat output from 20,000 kW to 50,000 kW and electrical output from 5000 to 6250 kW (restricted to this value by the size of the generator) but without withdrawing the control rods further than originally intended. The water boils at a higher rate and flows faster with natural circulation and without pump assistance. It is believed that output could be increased to 100 MW thermal and 25 MW electrical which would result in a reduction of power costs from 52 to 20 mills/kWh.

Power Reactors Fuelled with "Dry" Uranium Oxide. H. de Bruyn, B. L. A. Van der Schee and J. J. Went. *Combust. Boil. Ho. Nucl. Rev.* 1957, **11** (Dec.), 559–61.

Studies made by a Dutch group on the use of a flow of uranium particles through a homogeneous, a quasi-heterogeneous and a heterogeneous reactor have indicated that the last type would have several attractive qualities: (1) Producing high temperature steam without need to pressurize the reactor; (2) use of natural uranium in oxide form; (3) high conversion ratio and possibility of breeding of thorium oxide.

Analysis and Testing, Research

Precision Calorimetric Measurement. J. R. Pattison. *Research* 1958, **11** (May) 192–202.

A review of apparatus and techniques employed.

Determination of Pulverizing Properties of Coal in an OR-VTI Apparatus. P. I. Kiselev and L. N. Kondrat'eva. *Teploenergetika* 1958, (Jan.) 25–8 (In Russian).

The apparatus and method are described and results are given. The apparatus comprises a cylindrical chamber 22.4 mm in diameter, into which a 2.5 g prepared sample of coal is placed. Test pulverization is carried out by dropping a rod down a guide tube on to the sample from a predetermined height.

From Fuel Abstracts 1958, **23** (June) 5593.

Testing of Solid Fuels. Determination of the Composition of Fuel Ash. W. Radmacher and W. Schmitz. *BrennstChemie* 1958, **39** (June) 164–72 (In German).

The method proposed as standards for the determination of the following substances are described in detail: 1. SiO_2; 2. Al_2O_3; 3. TiO_2; 4. Fe_2O_3; 5. MgO; 6. Na_2O, K_2O.

Measurement of Steam Content in Flow. N. F. Ragulin. *Teploenergetika* 1958, **5** (Feb.) 51–5 (In Russian).

Two new systems, using hygrometers, for measuring the steam content in a steam water stream, are examined and test results considered. The first is based on separation of the mixture and measurement of the dynamic pressures of the mixture and of the dry saturated steam. The second, a non-separating system, consists of a pipe along which the mixture flows and into which a delivery tube and disc are fitted. Assuming absence of phase change, the water is evenly spread over the pipe cross-section and the disc measures only the output of dry saturated steam and is reckoned as for a single-phase system. Of the two systems the non-separating is simpler and better. In fact, in the measurement of steam content under unsteady conditions, the presence of a separator and hydraulic seal brings about distortion.

From C.E.G.B. Digest 1958, **10** (July 5) 1670.

Annual Report 1957. Council of the British Coal Utilisation Research Association. *Leatherhead* 1958, 62 pp.

The report outlines research carried out in the following fields: 1. Domestic heating (small-pipe central heating, small chain-grate stokers, room heating efficiency, smoke and sulphur from open fires); 2. Firing of industrial boilers (coking and chain-grate stokers, automatic stoker control, dust and grit emission, heat transfer and deposits in shell boilers); 3. Deposits and corrosion in water-tube boilers (corrosion of air heater and other cold surfaces, release of chlorine, alkalis and sulphur during combustion, instrumentation); 4. Gas producer practice; 5. Gasification (pressure gasification with slag tapping and quenching, tuyere and hearth design, viscosity of slag); 6. Basic research (carbonization and coking mechanism, carbonaceous products, solvent extraction, halogenation, physics and chemistry of coal); 7. General (cyclone combustion, pulverizers, cone mill, heat transfer, dropwise condensation, instruments and techniques). Details are also given of reports, published papers, lectures given by staff, patents held, interim reports, committees, staff, members and accounts.

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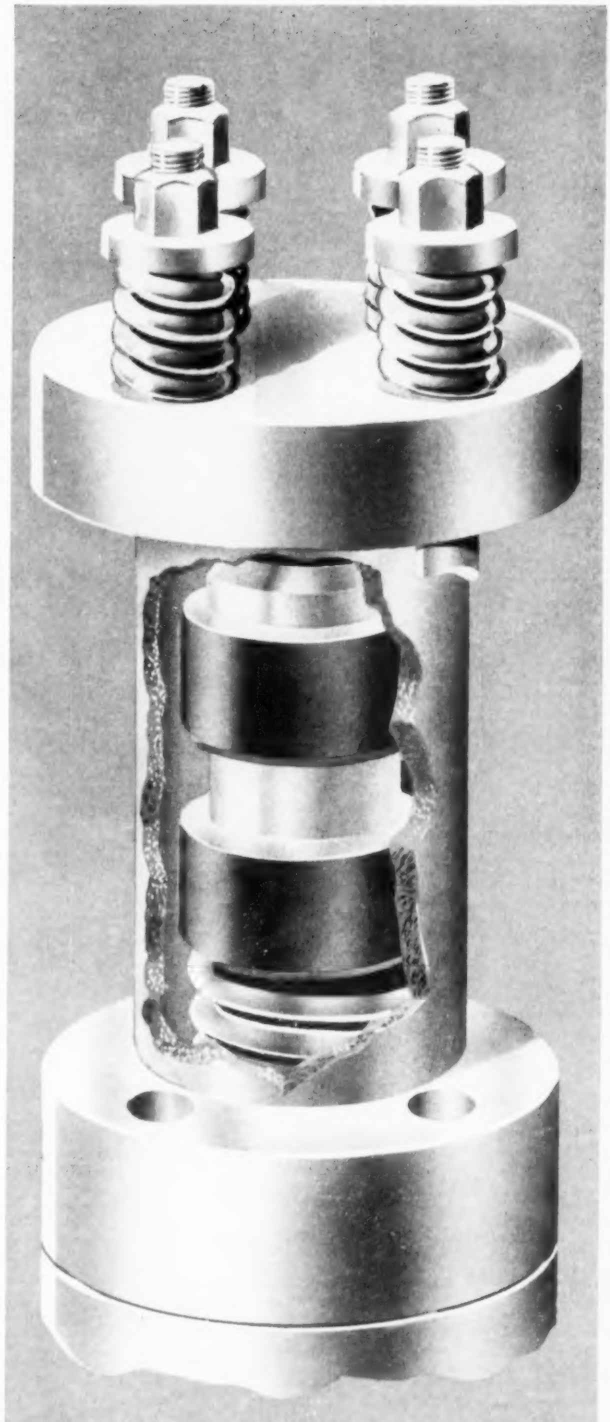

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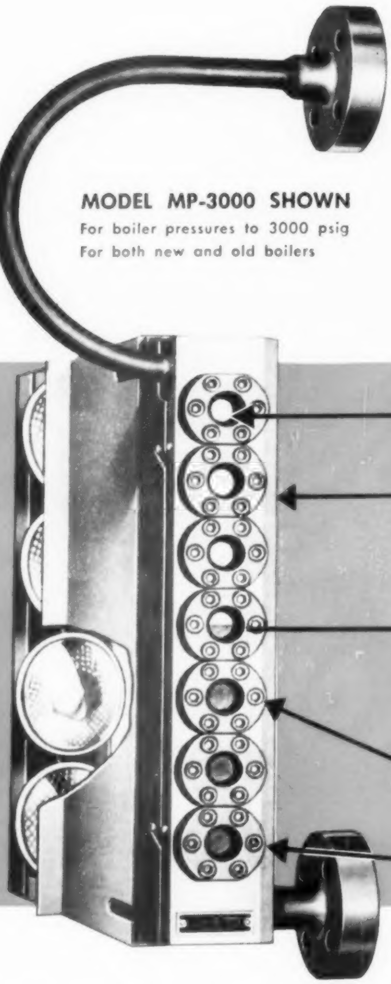

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